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Modelling wastewater aeration system and high-speed turbo blower for energy efficiency improvements

Master's thesis submitted for examination for the degree of Master of Science in Technology

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Tiivistelmä

Pyrkimyksissä parantaa jätevedenpuhdistamojen energiatehokkuutta, jäteveden ilmastuksen optimointi on usein esillä, johtuen sen korkeasta energiantarpeesta. Ilmastuksessa käytettyjen kompressorien ja prosessin ohjausstrategioiden kehitys tarjoaa mahdollisuuksia ilmastuksen energiankulutuksen vähentämiseksi.

Suurnopeus ilmastuskompressorit ovat varteenotettava vaihtoehto perinteisille kompressoritekniikoille, niiden energiatehokkuuden sekä muiden suurnopeusteknologian tarjoamien etujen ansiosta. Suurnopeustekniikka kuitenkin lisää kompressorien sähköistä mutkikkuutta, luoden erityisvaatimuksia laitevalmistajille. Tämän tutkimuksen tavoitteena on määrittää suurnopeuskompressorin tavanomainen toiminta jäteveden ilmastus prosessissa, jotta laitetason energiaoptimoinnin haasteet ja mahdollisuudet voitaisiin tunnistaa.

Tutkimus sisältää kirjallisuuskatsauksen pohjailmastusta käyttävän järjestelmän kokoonpanosta ja ohjausstrategioista, kompressoritekniikoista ja niihin liittyvistä taajuusmuuttajista sekä sähkömoottoreista. Tutkimuksen kokeellisessa osiossa Hermanninsaaren jätevedenpuhdistamon ilmastusjärjestelmä mallinnettiin suurnopeuskompressorien tyypillisten käyttömallien ja kuormitusten arvioimiseksi.

Mallilla tehtyjen simulointien tulokset osoittivat haasteita suurnopeuskompressorin sähkölaitteiden energiaoptimoinnille, mutta vastapaineen minimointi ohjausventtiilien ohjauksen optimoinnilla voisi tarjota suuria energiasäästöjä verrattain pienillä vähennyksillä vastapaineessa. Järjestelmille, jotka vaativat isompia kompressoriryhmiä, kompressorien ohjausstrategia tarjoaa enemmän mahdollisuuksia energiaoptimoinnille.

Avainsanat Jätevedenkäsittely, ilmastus, turbokompressori, energiatehokkuus, mallinnus

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Abstract

In efforts to improve wastewater treatment plants' energy efficiency, optimization of aeration process is often a point of interest due to its high energy intensity. Improvements in aeration blower technologies and process control strategies offer prospects for cutting down aeration energy consumption.

Alternative to traditional blower technologies is the high-speed turbo blower, which along with improved energy efficiency offers many fringe benefits, which makes it a worthy candidate for blower replacement projects. With the high-speed technology, blower's electrical complexity increases, creating some special considerations for the component manufacturers. This study attempts to characterize typical operation of a high-speed turbo blower in wastewater aeration process, in order to identify challenges and possibilities for energy efficiency improvements on equipment level.

Study includes literature review on diffused aeration systems structure and control strategies, blower technologies and associated variable frequency drives and electric motors. A model of the diffused aeration system and high-speed turbo blowers of the Hermanninsaari wastewater treatment plant is developed for assessing the typical operating patterns and loadings of high-speed turbo blowers.

Simulation results highlighted the challenges for energy efficiency optimization of high-speed turbo blower's electrical equipment, whereas minimizing the backpressure through valve control optimization algorithms, could offer large savings for relatively small reductions in the backpressure. For systems operating larger blower groups, blower operating strategy has more possibilities for energy optimization.

Keywords Wastewater treatment, aeration, high-speed turbo blower, energy efficiency, modelling

Foreword

Writing this thesis would not have been possible without the support from ABB Ltd, where I have had the pleasure of working for the last few years of my studies. I would like to thank the company for the opportunity to write my thesis on a topic that was every bit as interesting as it was challenging.

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Helsinki 22.04.2020

Lauri Pöyry

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Nomenclature

f	[Hz]	supply frequency
K_v	[Nm^3/h/bar]	valve flow coefficient
k	[-]	resistance factor
N	[rpm or Hz]	rotational speed
N_r	[rpm or Hz]	rotor speed
N_s	[rpm or Hz]	synchronous speed
n_p	[-]	pole number
P	[kW]	power
p	[kPa]	pressure
Q	[Nm^3/h]	air flow rate
S	[-]	slip

Abbreviations

AMB	Active Magnetic Bearing
BEP	Best Efficiency Point
DO	Dissolved Oxygen
MOV	Most Open Valve
NMSE	Normalized Mean Square Error
PLC	Programmable Logic Controller
PM	Permanent Magnet
UPS	Uninterruptible Power Supply
VFD	Variable Frequency Drive
WWTP	Wastewater Treatment Plant

1 Introduction

Increasing trends in energy prices and increasingly stringent limits for effluent quality motivate the wastewater treatment plants (WWTPs) for finding ways to improve their energy efficiency. For improving the plants total energy efficiency, optimizing the aeration process is often a point of interest due to its high energy consumption. Wastewater aeration is a very energy intensive process which can make up 45-75% of the WWTPs total energy demand (Rosso et al. 2008). The energy itself is consumed by the blowers that produce the air flow for the process and they are often found as the source of excessive energy usage in the aeration system. This is why improvements in the blower technologies and process control can yield substantial savings in plants energy costs. (Bell & Abel 2011)

Relatively new alternative to more conventional blower technologies is the high-speed turbo blower, which is often associated with improved energy efficiency. Case studies by Schilling and Turriciano (2011) showed 32-38% energy savings with high-speed alternatives while Bell and Abel (2011) stated that high-speed units can easily yield 35% reduction in energy consumption. Because of their operating principle, high-speed units have increased electrical complexity and they are inherently speed controlled as they always require variable frequency drive (VFD) to operate.

In contrast to many industrial processes, the loading of the treatment plant varies continuously and uncontrollably. Daily fluctuations in population activity cause diurnal variation, temperature changes seasonally and the plant is occasionally subject to discrete events such as heavy rains and industrial slug loads. (Jenkins 2013) The variation in these disturbances is extremely large and their time scales vary from hours to months (Olsson 2005). These are the main factors that create challenges for energy efficiency optimization of the aeration process on both system and equipment level.

The aim of this study was to develop a model of a diffused aeration system with high-speed turbo blowers, based on actual plant and equipment manufacturer's data. This model is then used to assess the possibilities for equipment level energy optimization, by identifying the typical operating patterns and load points of the high-speed turbo blowers. Additionally, the model is used for identifying possibilities and challenges for energy efficiency optimization of diffused aeration system.

The study begins with a literature study which introduces the main components of diffused aeration system and basic control methods, to set up the framework in which the blowers operate. Different blower technologies and the basics of VFDs, electric motors and high-speed turbo blowers are presented, to establish their limitations and considerations related to energy efficiency. The experimental part of the study includes description of the developed aeration system model and the analysis of the simulation results. Through the simulation results, the application requirements for a VFD controlling a high-speed turbo blower are identified. Implications regarding the challenges and potential energy efficiency improvements are discussed.

2 Aeration system

The task of the wastewater aeration is to provide sufficient amount of oxygen for aerobic organisms that perform the removal of organic matter and nitrification in activated sludge process (Åmand et al. 2013). If there is not enough oxygen available, the microorganisms cannot metabolize organic matter or convert ammonia to nitrate. Prolonged operation without sufficient amount of oxygen can also lead to biomass die-off (Harja et al. 2016). Although the minimum required dissolved oxygen (DO) concentration is plant specific, in general, operation above concentration of 1 mg/l is adequate. On the other hand, increasing the amount of dissolved oxygen above the minimum requirement does not accelerate the removal of organic matter. Potential nitrification rate is also limited, and operation above DO concentration of 3 mg/l does not increase the ammonia conversion. The oxygen transfer efficiency decreases exponentially with higher DO levels, so it is more energy efficient to operate at the minimum DO level that meets the performance requirements. (Jenkins 2013) Operating at excessive DO levels can also harm the process, as it may cause the degradation of sludge quality (Harja et al. 2016). Apart from supplying oxygen for the process, aeration also needs to provide sufficient mixing to keep the biomass in suspension, which adds another constraint for the minimum feasible level of aeration (Åmand et al. 2013).

Aeration system can be divided into three different sections, which are illustrated in the figure 1. First is the air flow generation, which is done by using blowers that supply the system with pressurized air, second is the distribution system for that air flow, which consists of piping, control valves and diffusers. Last is the aeration control system, which consists of the feedback devices and the control algorithms for control valves and blowers. For optimal operation, all three parts need to be working well together, as the improper design or operation of one of the aspects influences the rest of the system, leading to either decreased process performance or energy efficiency. Well-designed aeration system can provide substantial energy savings, while maintaining stable operation of the biological process. (Gray et al. 2011)

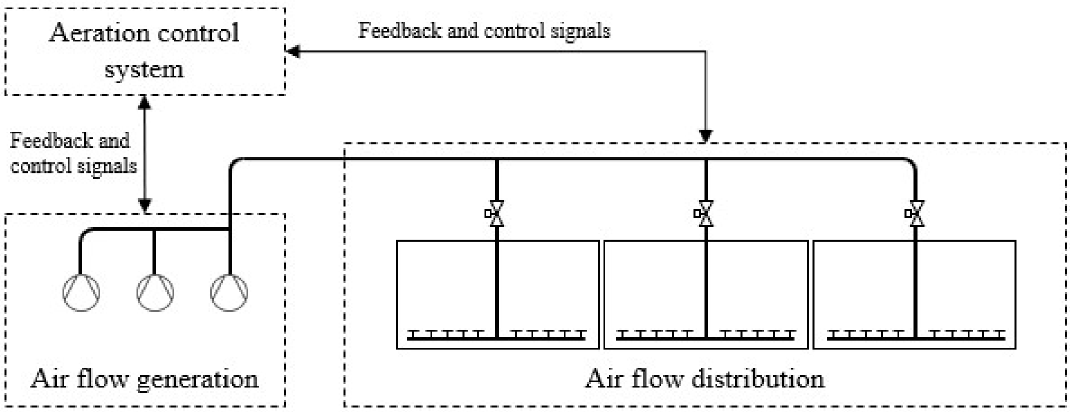

Figure 1. Schematic of diffused aeration system

2.1 Air flow generation

Air flow required to meet the target DO level, is generated by using blowers. The blower system is the largest single users of energy at WWTPs, so their efficient operation is of great interest to the plants (Åmand et al. 2013). For existing plants, the blowers are frequently the source of excessive energy consumption and the blower replacement is often identified as high priority investment for improving plants energy efficiency (Bell & Abel 2011).

There are multiple different types of blowers, but they all consist of rotating lobes, impellers or screws that pull in outside air and force it through the air flow distribution system. Categorization between the different blower types is made by their method of producing the air flow. Two main types are the positive displacement and centrifugal blowers, which can be further categorized into different subgroups by the type of their configuration. Due to different characteristics and capabilities of different blower technologies, the suitability of the blower type depends on the plant design and its aeration requirements. Typically, positive displacement blowers are suitable only for small plants, while centrifugal blowers have varying configurations that can be used in both small and large WWTP's. (Spellman 2013) Typical efficiencies and turndowns for different blower types are shown in the table 1. Different blower technologies are further discussed in the chapter 3.

Table 1. Typical blower efficiencies and turndowns (EPA 2010).

Blower type	Nominal efficiency (%)	Nominal turndown (%)
Positive displacement (variable speed)	45-65	50
Multi-stage centrifugal (inlet throttled)	50-70	60
Multi-stage centrifugal (variable speed)	60-70	50
Single-stage centrifugal, integrally geared (dual point control)	70-80	45
Single-stage centrifugal, gearless (high-speed turbo)	70-80	50

As there are large variations in the WWTPs loading, the flexibility of the blower system is important for the optimal operation of the plant (Åmand et al. 2013). Important parameter apart from blower's total efficiency, is the turndown, which describes the operation range of the blower. It is the ratio of the minimum flow to the maximum flow, that blower or blower system can safely provide. The turndown capabilities of a single blower unit are limited by the mechanical constraints of the blower design and the blower motors characteristics. The turndown of a single blower is around 50%, but this varies significantly, depending on the blower's type, design and the control method used for modulating the air flow. The aeration control system tuning may also affect the turndown, as the instability or poor control precision may prevent the blowers from operating at their full theoretical range. (Jenkins 2013)

In addition to turndown capability of single blower unit, turndown can be achieved by different configurations of the blower system. It is not uncommon to have 8:1 ratio between aeration demand of the worst-case scenario and the minimum demand, which for optimal operation would require the blower system to be capable of operating with 12.5% of its design capacity. Generally, blower systems should be designed for at least 5:1 ratio between maximum and minimum demand, as providing the turndown for 8:1 ratio can be challenging. Most regulatory agencies also require redundancy in the blower design, which means that the blower system should be able to meet the worst-case demand with the largest unit out of service. Some of the common blower arrangements and their turndowns assuming the 50% turndown of a single unit are presented in figure 2. (Jenkins 2013)

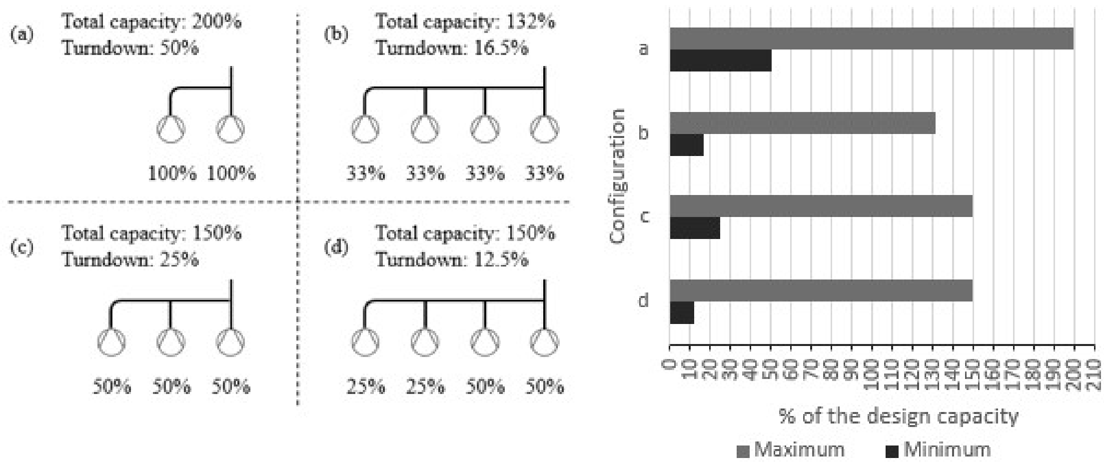

Figure 2. Common blower arrangements. Percentage for each blower represents how much of the total design air flow rate it can output at its nominal point. Total capacity is the percentage of the design air flow rate and turndown is the minimum percentage of the design air flow rate. Turndown for individual blower is assumed to be 50%.

From energy efficiency perspective, turndown is often more important than the efficiency of a single blower unit. This is because the limited operating range prevents the system from operating optimally during the periods with low air flow requirements, leading into over aeration and wasted energy. This is an issue that is common with the wastewater industry, as the design protocols often lead into oversized blower systems. Too conservative design assumptions that cause the oversizing of the blowers will lead into reduced control over the process, increased capital costs and energy consumption of the plant. (Rach-Williams et al. 2013)

2.2 Air flow distribution

The distribution system for the air flow generated by the blowers consists of pipelines, control valves and the diffusers. These components of the system define the system curve, which is based on the relation between air flow rate and pressure developed in the system. The total pressure consists of static and dynamic pressures. The static pressure is developed due to hydrostatic pressure over the diffusers and it can be assumed to remain constant. This represent the largest portion of the system pressure and as it is directly proportional to the diffuser submergence, the only way to affect it is by the tanks design. The dynamic pressure is caused by the air movement in the distribution system. The dynamic pressure in aeration system is affected by the design of the piping, the type and position of the control valves and the design of the diffuser grid and type of diffusers used. (Amerlinck et al. 2016) Simplification of different pressure components along the aeration system is shown in the figure 3.

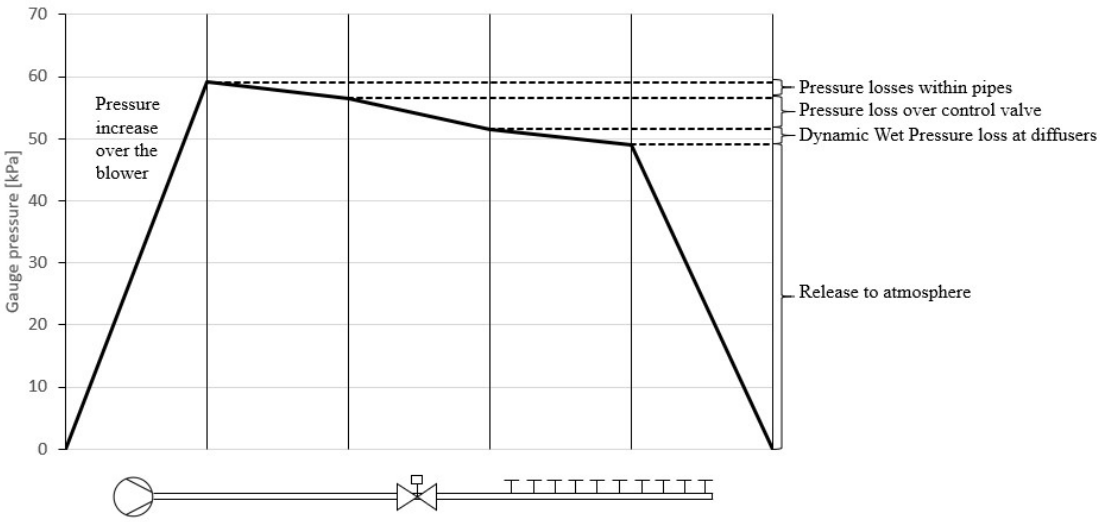

Figure 3. Example of different pressure components along the aeration system.

The factors affecting the losses within the pipeline are the diameter and total equivalent length of the pipes, air temperature, air flow rate and the material of the pipe (Amerlinck et al. 2016). The dimensioning of the piping is important as oversized pipelines can make the control of the air flow difficult, while using too small pipe diameters leads to excess pressure losses and consequently increase the blower systems energy requirements (Gray et al. 2011). Pressure losses within the pipeline are typically less than 10% of the total system pressure (Spellman 2013).

Control valves are used for managing the distribution of the air flow. They throttle the air flow to specific segments or isolate segments by closing off completely. As the throttling is about creating variable restriction to the air flow, there is a parasitic pressure loss over the control valve. This pressure loss increases the power requirements of the blowers, so the aeration control system should be minimizing the amount of throttling in order to improve the energy efficiency of the system. How much pressure loss is caused by the valve depends on the valve opening, air flow rate, properties of air and the valve specific characteristics. Typically butterfly valves are used because they are economical, they are suitable for both throttling and shutoff, and they are available in large variety materials. The valve specific characteristics are typically provided by the manufacturers and they are stated as K_v values for different valve openings. The K_v value represents the volume of water in m^3/h that will pass through the valve at a pressure drop of 1 bar and as the value is given for water, it needs to be adjusted for the air, before calculating the actual pressure drop. (Jenkins 2013)

The last part of the air distribution system are the diffusers, which most typically are fine pore diffusers, which produce small bubbles releasing air through small orifices or pores (Rosso et al. 2008). Example of fine-pore diffuser installation is shown in the figure 4. The pressure loss associated with the fine-pore diffuser systems is called dynamic wet pressure loss, which depends on the diffuser fouling, product specific characteristics and the diffuser flux. For a fixed number of diffusers used, increasing air flow increases the dynamic wet pressure loss. This component of the total system pressure can be affected by the diffuser selection, layout and periodic cleaning. (Amerlinck et al. 2016)

Figure 4. Example of installation of fine-pore diffusers (Henze 2008).

The distribution system described above defines the system curve, which intersects with the blower curve in one point only. This point expresses the only possible flow rate and pressure in that system, and it will be the operating point of the blowers. If there is a shift in the system curve due to changes in the dynamic pressure, blower control needs to be utilized for keeping the flow rate at the desired point. (Amerlinck et al. 2016) Example of system curve with two different valve positions is shown in the figure 5.

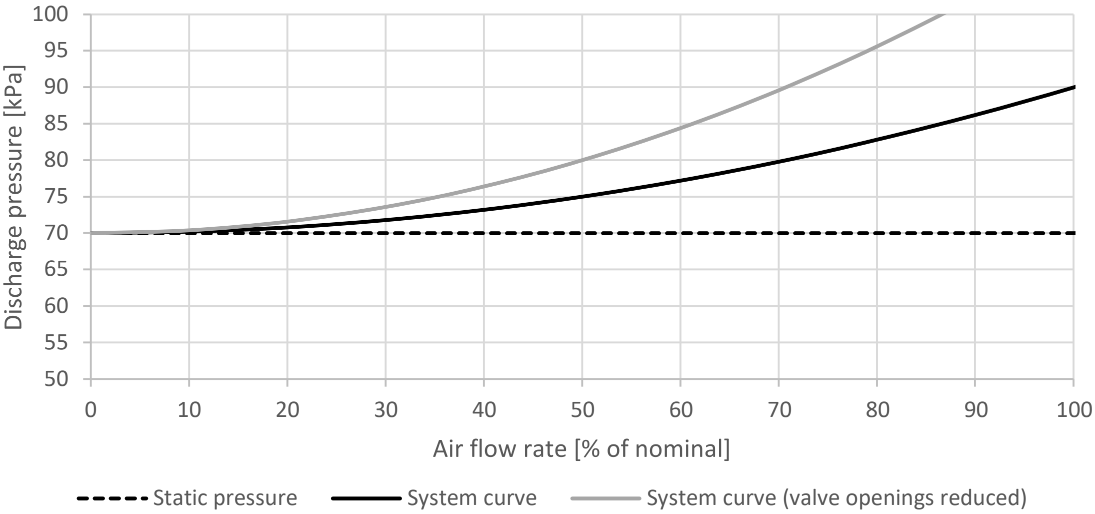

Figure 5. Effect of reducing valve opening on system curve.

2.3 Aeration control

The goal of aeration control is to keep the DO levels in the aerated reactors at the desired setpoints. The oxygen requirement of the process is dynamic, due to variations in the plants loading, which means that in order to sustain the desired DO setpoint, air flow to the system needs to be regulated (Gray et al. 2011). The automated control of the aeration improves the process supervision and enables operation closer to the limits set by the requirements for the effluent quality (Åmand et al. 2013).

Generally, aeration control systems utilize two autonomous feedback loops, first loop being for the DO concentration and the second one being the pressure control loop. These loops use the positions of the valves and the variable controlling the air flow rate from the blowers as manipulated variables. The latter depends on the blower control method used, and it can be for example the rotational speed of the blower. (Piotrowski et al. 2004) Additional control loops may be used, if the system adjusts the DO setpoint based on the influent loading or if the control valves use separate controllers for controlling air flow of individual zones. Typical pressure control loop can also be removed, if the system uses direct air flow control for managing the air flow rate from the blowers (Spellman 2013).

2.3.1 DO control

The automated DO control strategies can vary from simple on/off or setpoint control to complex systems utilizing model-based algorithms. Common strategy is to use cascade control, where first control loop outputs control signal for basin air flow control according to feedback from the DO probes. If the measured DO deviates from its setpoint, the air flow rate to the basin needs to be adjusted. This control loop can directly adjust the control valve position or in some cases separate flow meters and flow controllers are used for determining the valve opening. (Spellman 2013) Typical arrangement for DO control with separate flow controller is shown in the figure 6. From the control perspective, the use of separate flow controllers for the control valves is beneficial, as the faster acting inner control loop for air flow improves the rejection of internal disturbances and linearizes the non-linear valve characteristics (Vrečko et al. 2014).

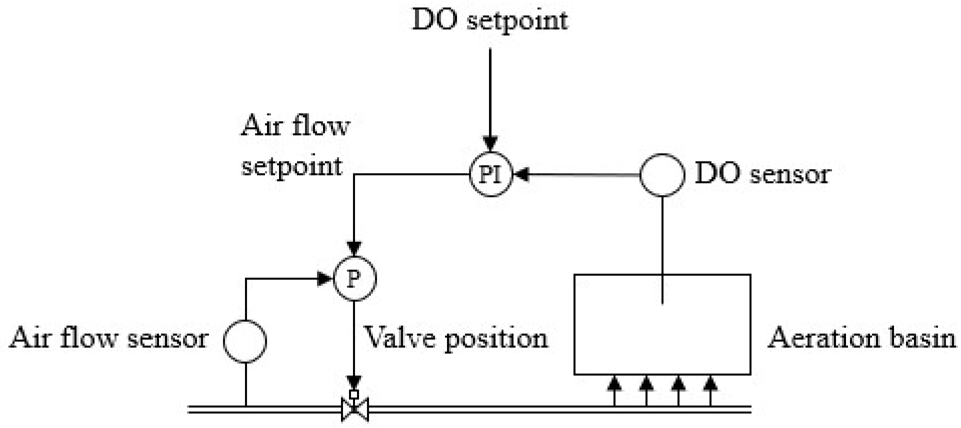

Figure 6. Typical control strategy for DO control.

The DO setpoint for the aeration control is either set manually by the operator or through optimization algorithms (Harja et al. 2016). For example, the optimization algorithms can use ammonia measurement or process models to estimate the optimal setpoint for the DO. The ammonia measurement-based systems include both feedback, feedforward and the feedforward-feedback controls. For ammonia feedback control, the ammonia measurement is taken from the outlet of the aerated reactors and for the feedforward control the incoming ammonia load is measured. The feedforward control will react faster to the disturbances, but as its control actions are based on predictions about the impacts of the disturbance, the performance of the control depends on the quality of the model used for relating the ammonia loading to optimal DO setpoint. The feedforward-feedback control works similarly to the feedforward controller, but in addition it uses feedback signal from the outlet for making corrections to its control actions. (Åmand et al. 2013) Both ammonia feedback and feedforward control loops are shown in the figure 7.

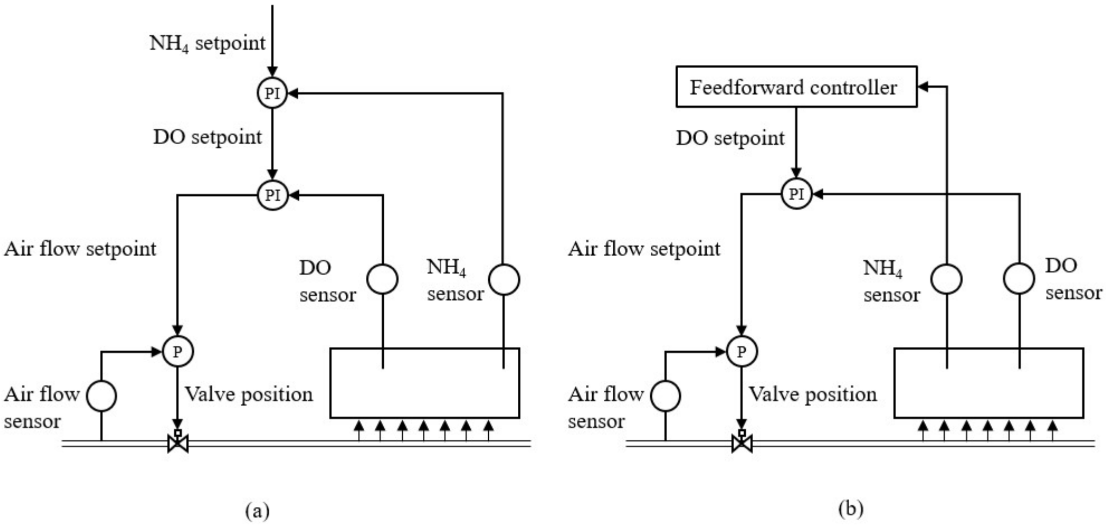

Figure 7. Ammonia based feedback (a) and feedforward (b) control strategies for DO control.

More advanced control systems use model-based controllers, that work on top of the conventional feedback systems. The control-law of these controllers is based on a process model, which can be either black-box, without any relation to theorical characteristics of the process or it can be white-box which works based on assumptions of the theoretical process dynamics. The models are then used for finding the most optimal outputs for the process control. (Åmand et al. 2013) Systems utilizing advanced control strategies is highly dependent on the performance of the model and how well it captures the actual dynamics of the process.

2.3.2 Pressure control

The pressure control is typically used in medium and large sized plants, where blowers supply air to a common air rail, from which the control valves distribute the air to individual zones of the process (Alex et al. 2002). The function of pressure controller is to maintain constant pressure at the common rail, by modulating the air flow rate of the blowers. The method of controlling the air flow rate indirectly by using the pressure control, should prevent the disturbances from individual valve operation from affecting the rest of the system. The setpoint for the pressure is either kept constant, or it is minimized by using Most-Open-Valve logic. (Serralta et al. 2002) The tuning of the pressure controller is important, as improper parametrization can lead to large fluctuations in the pressure at the common rail and deteriorated control accuracy of the total air flow (Vrečko et al. 2014). An example of the pressure control loop is shown in the figure 8.

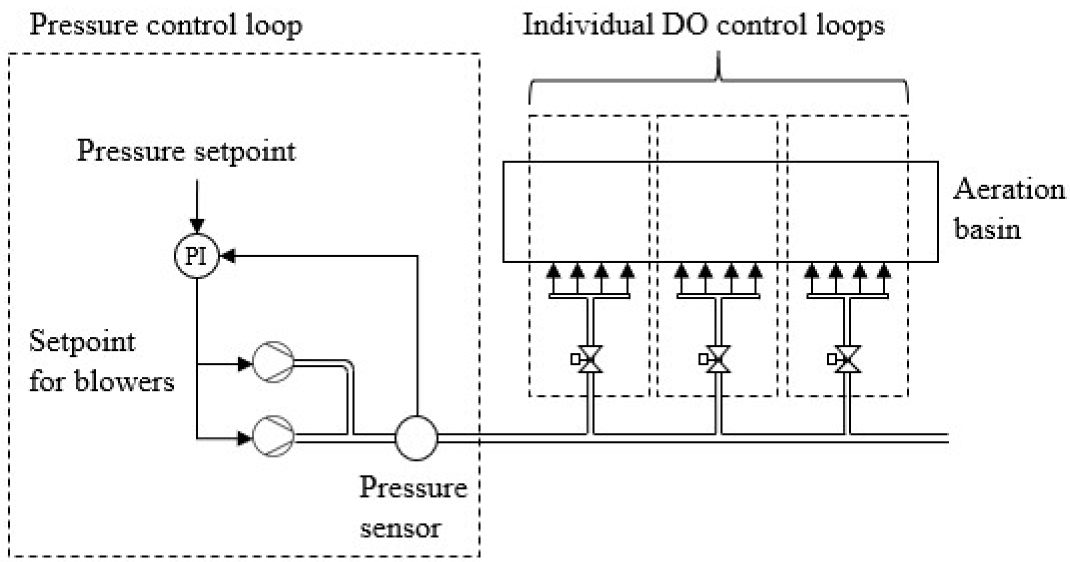

Figure 8. Example of pressure control loop and its physical location in the aeration system.

In the traditional pressure-controlled aeration systems, hunting of the blowers and the control valves is often observed. The hunting occurs when pressure and valve control loops respond to each other's control actions instead of the actual process changes (Gray et al. 2011). The underlying reason for this hunting is that the aeration control system does not recognize the limited operational ranges of the blowers and the fact that with the constant pressure system, the changes in individual valve position will influence the air flow rates of the other sections rather than directly affecting the blower systems output (Henze 2008). The latter can also be deduced from the figure 8, which shows that the control loops for pressure and DO are completely separated. If one of the control valves is opened, the air flow through the other valves will be reduced and in turn they need to also increase their openings. The total air flow range of the blower system may have gaps, because of the limited turndown capabilities of a single blower, which leads to that the starting an additional blower may exceed the actual air flow requirement. Example of the total air flow range for two blowers, with 60% turndown and maximum air flow rate of 2500 Nm^3/h is shown in the figure 9.

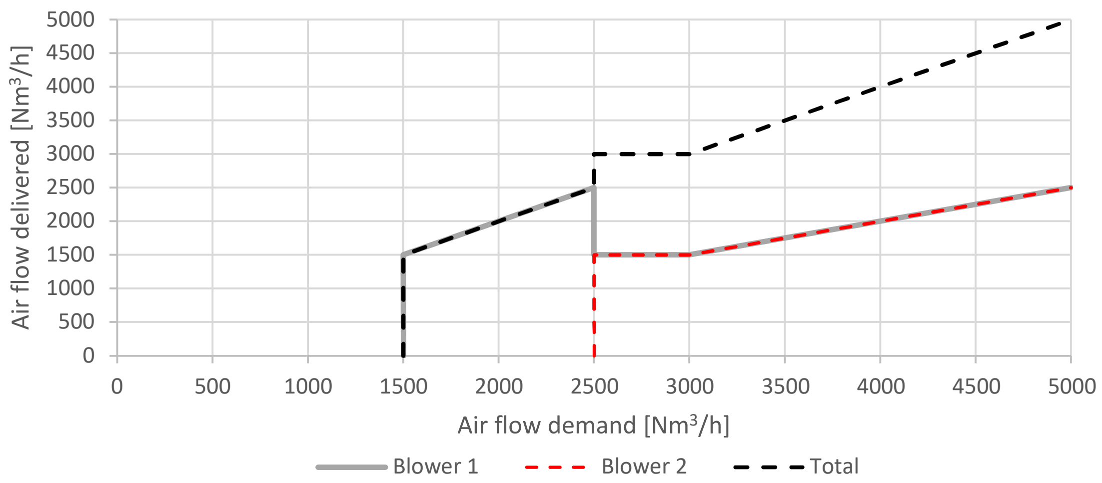

Figure 9. Limited total air flow range of two blowers with 60% turndown and maximum air flow rate of 2500 Nm^3/h.

From the figure 9 we can see that air flow demand exceeding 2500 Nm^3/h will require starting of additional blower, but it is not possible to supply air flow rates less than 3000 Nm^3/h, if the minimum flow rate for a single unit is limited to 1500 Nm^3/h. Eventually, one of the valves will start to close, forcing more air flow through the rest of the system, which will lead into closing down of the other valves. As the valves are closing, the system backpressure will increase, leading into shutting down of the previously started blower. This cycle will then start from the beginning as the DO level in one of the controlled zones starts to decline. The hunting will result in excessive energy consumption and increased wear of the blowers. (Henze 2008)

2.3.3 Direct flow control

Direct flow control is an alternative to the pressure control, where the total air flow from the blowers is directly controlled to match the process demand (Åmand et al. 2013). Directly controlling the air flow leads to more simple control structure and improved operation of the blowers. The use of additional pressure control loop for indirectly controlling the air flow can cause instabilities especially at smaller plants, such as previously described hunting. (Spellman 2013). In addition to decreased efficiency, hunting is suboptimal from the control perspective and it may lead to premature failure of the actuators. (Gray et al. 2011) Example of direct flow control of the blowers is shown in the figure 10.

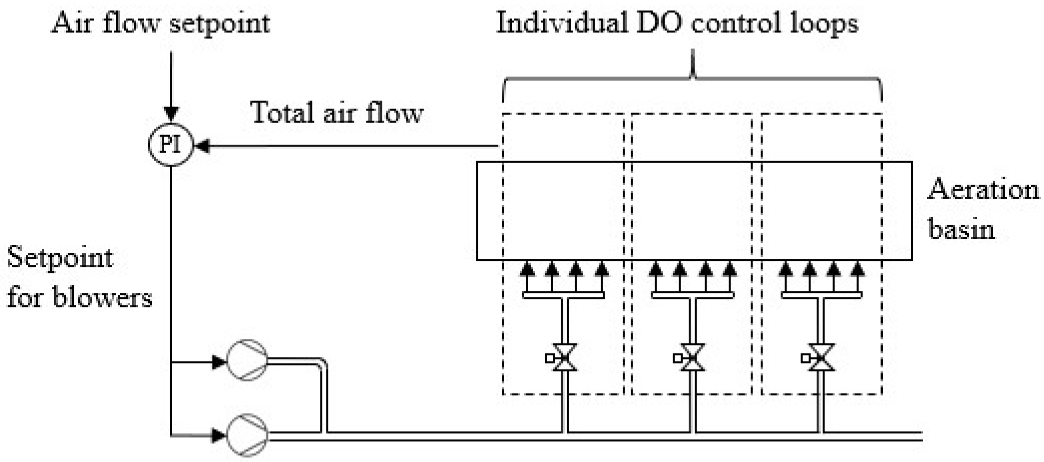

Figure 10. Direct flow control of the blowers.

The operation at the lowest possible pressure is important as excess pressure greatly increases the power consumption of the blowers (Vrečko et al. 2014). Conventional pressure-based systems waste energy, if the pressure setpoint is set too high. The pressure setpoint can be optimized but using the pressure as controlled variable is not as optimal as using air flow. In practice, the direct air flow control has challenges related to the inaccuracy of the flow measurements. Accurate flow measurement requires sufficient length of straight pipe, so that the velocity profile of the flow is uniform. The lack of sufficient straight sections is the most common problem for getting reliable flow feedback from the system Another problem for flow measurements is caused by the control valves, which by their movement shift the velocity profile of the flow and cause false readings. This is why the flow meter should always be installed upstream of the control valves if possible. (Jenkins 2013) The issues related to acquiring reliable feedback on the air flow rate should be considered during the design of the piping and selection of the measuring equipment.

2.3.4 Most-Open-Valve logic

Most-open-valve (MOV) is a control logic that is used for minimizing the excess dynamic pressure losses over the control valves. Proper operation of the logic results in a minimum required discharge pressure of the blowers and consequently the power requirements for producing the air flow are minimized. It should be noted that it is not a part of the DO or blower flow control, but a separate logic to modify the other control loops. Initially it was developed for the pressure-controlled systems, but later it was modified for systems utilizing direct flow control. (Jenkins 2013) For both systems, the MOV-logic ensures that the valve controlling the zone with the highest aeration demand, is kept at its maximum position, eventually minimizing the throttling required for distributing the air flow into different zones of the system (Spellman 2013). MOV logic for both pressure and flow based systems are shown in the figure 11.

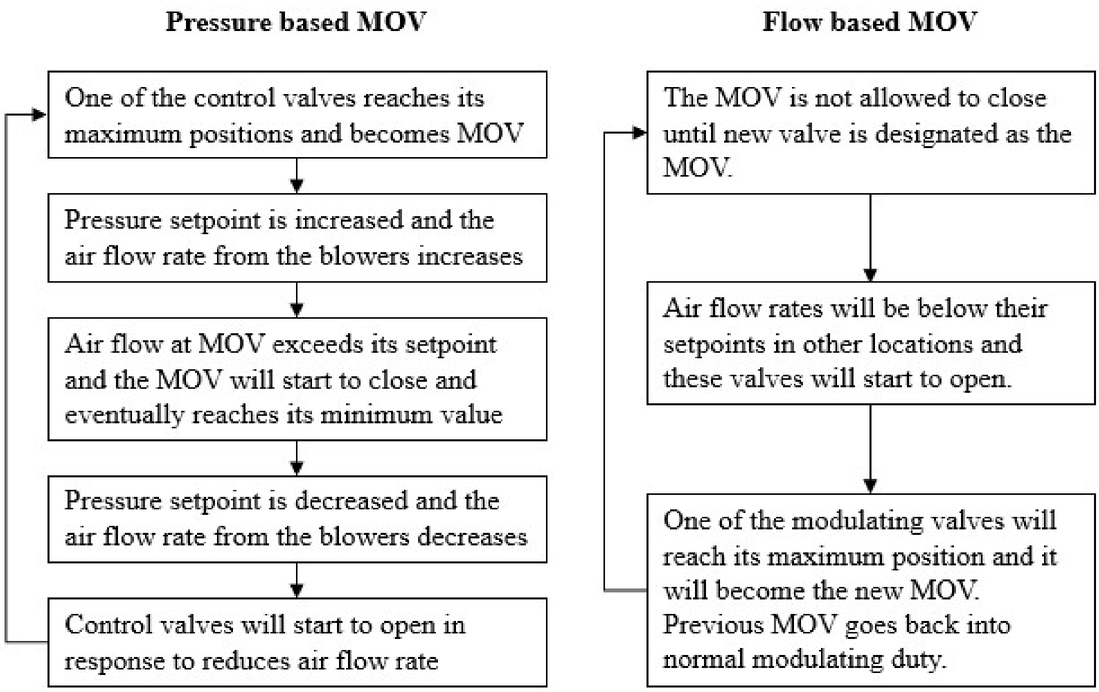

Figure 11. MOV logics for pressure and flow based systems (adapted from Jenkins 2013)

3 Blower technologies

3.1 Blower types and characteristics

For every blower that operates at a constant rotational speed, discharge flow rate decreases as a function of the discharge pressure. This relation between blower's discharge flow rate and pressure defines the blower's characteristic curve, which depends on the blower's type and design. The blower type has a large influence on this relation, as due to different operating principles, positive displacement and centrifugal blowers have significantly different characteristic curves. In addition, the efficiency of the blower varies non-linearly and has its optimum point which is known as the best efficiency point (BEP). (Amerlinck et al. 2016) Because of their characteristics, positive displacement blowers have been categorized as a constant flow, variable pressure devices, whereas centrifugal blowers have been considered as a constant pressure, variable flow devices. These descriptions are true only in a sense that they describe the shape of the characteristic curve, but due to technological advancements such as variable speed control, the operating range has expanded for both blower types. Still, the operating range of the blower is dependent on the blower type and design, as the operation of positive displacement blower is limited mainly by the characteristics of the blower motor, whereas for the centrifugal blower the aerodynamic capability of the impeller and its housing is typically the limiting factor for the safe operation. (Jenkins 2013) Typical blower curves for positive displacement and centrifugal blower are illustrated in a figure 12.

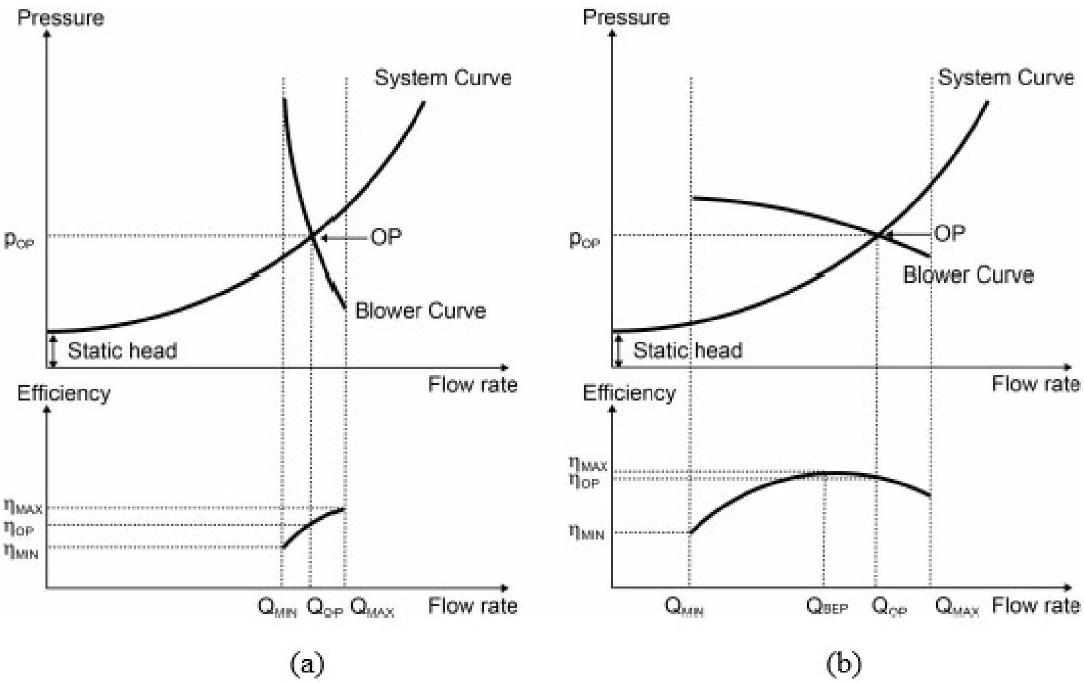

Figure 12. Typical blower and efficiency curves for (a) positive displacement blower and (b) centrifugal blower (Amerlinck 2016).

The blower characteristic curve, together with the system curve defined by the aeration distribution system components described in the chapter 2, will determine the operating point of the blower. For controlling the flow rate from the blowers, either the blower curve or the system curve needs to be modified, in order to adjust the operating point. (Amerlinck et al. 2016) Common blower control strategies include inlet throttling, variable speed, variable inlet guide vanes and variable discharge diffusers (Spellman 2013). The effects of inlet throttling and variable speed are shown in the figure 13.

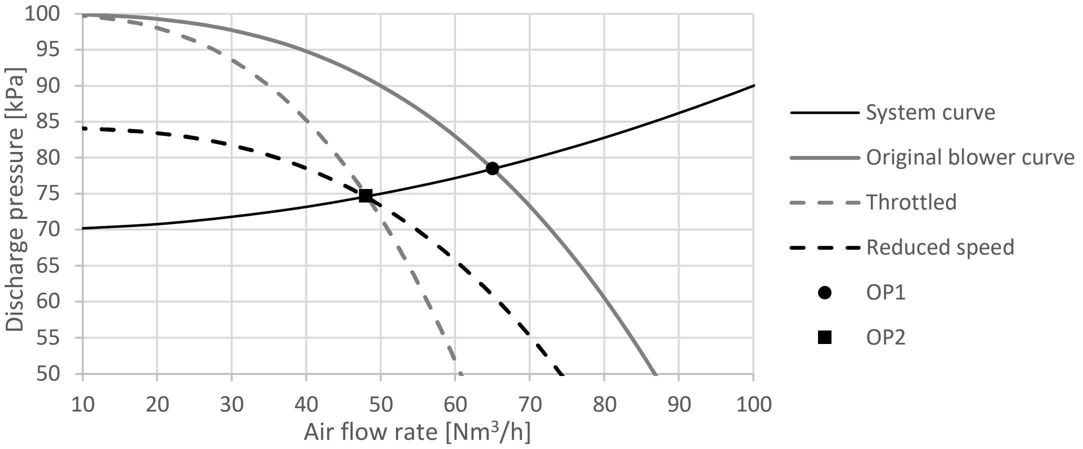

Figure 13. Effects of inlet throttling and variable speed on the blower curve and operating point.

Because of the different characteristics between the blower technologies and the varying plant sizes and aeration system designs, blower configuration and blower selection are highly depended on the system at hand. Traditional technologies used in WWTPs are the different positive displacement blowers, multistage centrifugal and single-stage integrally geared centrifugal blowers (Rohrbacher et al 2010). Relatively new technology is the single-stage direct drive centrifugal blower, which is often referred to as a high-speed turbo blower, which can offer substantial energy savings in comparison to more traditional blower technologies (Bell & Abel 2011).

3.1.1 Positive displacement blowers

Positive displacement blowers use two counter rotating shafts equipped with impellers to push fixed volumes of air through the blower. By the type of impellers used, positive displacement blowers can be separated into two main types, both of which are widely used in wastewater aeration systems – rotary lobe and screw blowers. (Jenkins 2013) An example of the operating sequence of a twin lobe positive displacement blower is shown in the figure 14. Positive displacement blowers' discrete compression process makes them typically the least energy efficient of the blowers, but due to their low capital cost, they are economical option for the smaller treatment plants (Spellman 2013). The operation of the positive displacement blowers is quite noisy, and they often require inlet and outlet silencers, which increase the pressure loss over the blower and therefore increase the blower's pressure and power requirements. On the other hand, as positive displacement blower's operation is mainly limited by the motor's performance, they can have significant turndown capabilities. (Rohrbacher et al 2010) Positive displacement blowers will not be discussed in detail, as the emphasis of this study is on the centrifugal blower types.

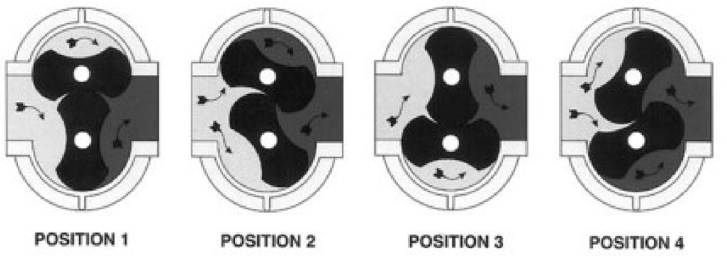

Figure 14. Operating sequence of a twin lobe positive displacement blower. The light grey areas within the blower casing indicate the inlet air and the darker areas indicate the outlet air, which is at the line pressure. (Giampaolo 2010)

3.1.2 Centrifugal blowers

The operation of centrifugal blowers is based on the continuous radial acceleration of the air, by using either single impeller or a sequence of impellers connected on a single rotating shaft (Schilling & Turriciano 2011). The rotating impeller intakes air axially at the impeller eye, after which the air is flung outward by the impeller vanes. The high velocity air leaving the impeller is then decelerated in the diffuser section of the blower. This deceleration causes some of the flows velocity pressure to convert into a static pressure. The conversion of velocity to static pressure continues as the air flows through the volute before being collected into the discharge pipe of the blower. Due to low density of air, imparting enough kinetic energy for creating the required discharge pressure can be challenging. This is why centrifugal blowers need either multiple consecutive stages or high rotational speeds for producing sufficient pressure. (Jenkins 2013) An example of a centrifugal blower is shown in the figure 15. The single stage units are typically rated for 49-62 kPa discharge pressures, while multiple blower stages enable the operation with higher discharge pressure (Bell & Abel 2011).

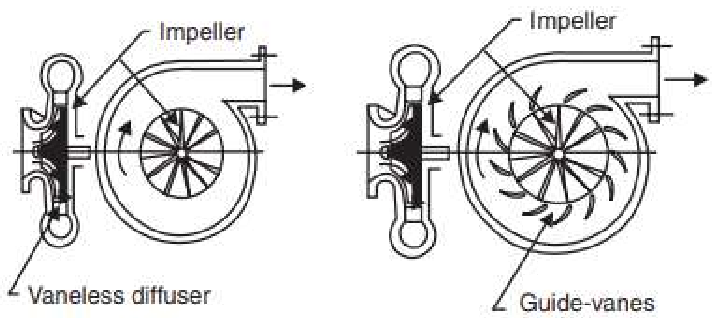

Figure 15. Side and top cross-sections of a centrifugal blower with and without the diffuser guide vanes. (Kadambi & Prasad 2015)

Multistage units produce the discharge pressure incrementally by increasing the pressure in several stages. These units consist of series of impellers, connected on a common shaft, that rotates typically at the rated speed of the blower's motor, which depends on the motor type and the network in which the motor is connected. For example, two pole AC motor in 60 Hz network, will run at around 3600 rpm. The advantage of multistage units is their higher energy efficiency compared to positive displacement blowers and their lower capital cost compared to the single stage units. (Spellman 2013) Cross-section of multistage centrifugal blower is presented in figure 16a.

Single stage units were introduced to the wastewater industry in early 1980s (Bell & Abel 2011). Instead of producing the pressure in multiple stages, single stage units rely on higher rotational speed, which enables the production of sufficient discharge pressure with a single impeller. Traditionally, the higher rotational speeds are reached by using speed increasing gears and these units are more specifically categorized as the single stage integrally geared centrifugal blowers, to differentiate them from the direct drive units. The rotational speeds for integrally geared units typically range from 10 000 to 14 000 rpm. They are more expensive than their multistage alternatives, but they can offer improved energy efficiency and turndown. (Spellman 2013) Single stage integrally geared units typically use variable inlet guide vanes, possibly together with the variable discharge diffusers for modulating the air flow (Rohrbacher et al. 2010). Cross-section of single stage integrally geared centrifugal blower is presented in figure 16b.

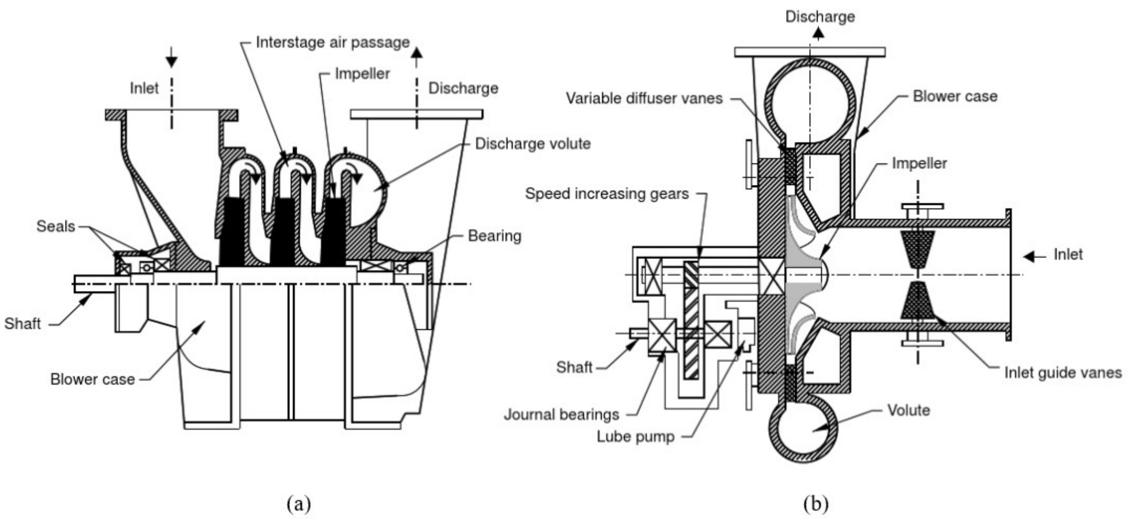

Figure 16. Cross-sections of multistage (a) and integrally geared (b) centrifugal blowers (Jenkins 2013)

Newer alternative for the traditional centrifugal blowers is the high-speed turbo blower. They are essentially single stage direct drive centrifugal blowers, which instead of using speed increasing gears, use high-speed motor with friction free bearings for reaching high rotational speeds (Schilling & Turriciano 2011). Their benefits include small footprint, high efficiency and good turndown. Although they are typically more expensive than the multistage centrifugal or positive displacement blowers, their capital costs are generally less than of the integrally geared alternatives. (Spellman 2013) Rohrbacher et al. (2010) states that the high-speed units typically operate between 15 000 and 25 000 rpm. As the high-speed motors rated powers decrease with the increasing speeds, smaller units might run at over 40 000 rpm. Because of the direct drive and high rotational speeds, high-speed units always require VFD to operate (Binder & Schneider 2007). The VFD also provides the adjustable speed control, which is the common control method of the high-speed units. Conventional geared system and high-speed system are shown in the figure 17.

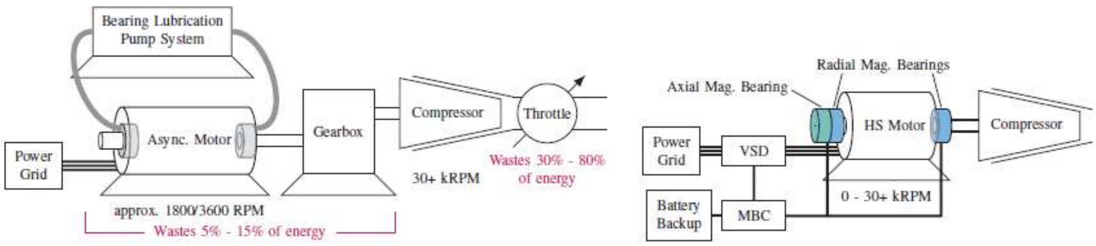

Figure 17. Conventional industrial motor system (left) and high-speed system utilizing magnetic bearings (right). VSD is the variable speed drive and the MCB is the Magnetic bearing Controller. (adapted from Severson & Mohan 2017)

The operational range of a centrifugal blower is limited by the choking of the flow at the high air flow rates and aerodynamic instability known as the surge at low flow rates (Gravdahl & Egeland 1999). In addition to the choking, blowers maximum air flow rates can also be limited by the blower motors power rating (Jenkins 2013). Example of centrifugal blower's operational limits is illustrated in the figure 18.

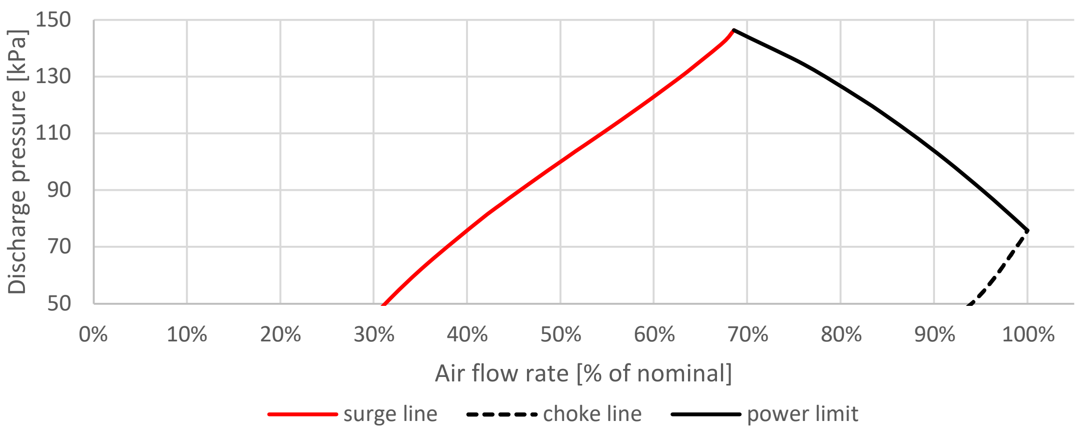

Figure 18. Operating limits of centrifugal blower.

The choking of the flow occurs when the air reaches sonic velocity at some part of the blower and the flow rate will become velocity limited. As a blower operating at a constant speed approaches the choke point, the pressure produced will drop as the flow approaches sonic velocity. This choke point is also known as the stonewall after which no additional flow can be produced. The choke points in different rotational speeds define the choke line, which will be the maximum limit of the blowers useful operating range. (Gravdahl & Egeland 1999) Choking of the flow is not as severe as the blower surge, but it still is undesirable condition to operate in.

For centrifugal blowers, the operation is limited at low flow rates by the unstable surge condition. Surge is a phenomenon, in which the direction of the air flow reverses in cycles with a frequency of around 2 Hz. (Hall 2018) During the surge cycle, violent flow oscillations subject the blower's impeller and casing to high mechanical stress and temperatures, potentially causing extensive damages (Yoon et al. 2010). Typical surge cycle is depicted in the figure 19.

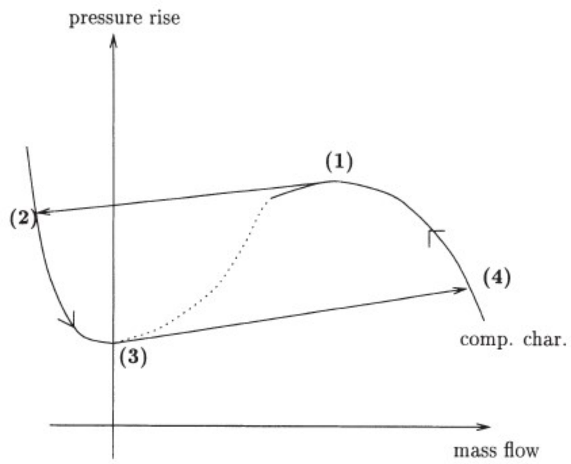

Figure 19. Typical surge cycle. The cycle starts from point 1 and shifts to point 2 with negative mass flow rate. The reversed mass flow rate decreases to near zero flow at point 3 after which the direction shifts back to normal at point 4, from which the operating point moves along the characteristic curve back to point 1 and the cycle starts from the beginning. (Gravdahl & Egeland 1999)

Although surge can occur in any type of centrifugal blower, single stage units are more sensitive to surge than their multistage alternatives. Small multistage units can operate under surge conditions for minutes, whereas surge in high-speed turbo blowers can lead to catastrophic failure in a matter of seconds. (Jenkins 2013) Due to the extent of potential damages, different methods for avoiding and controlling surge have been developed.

The surge line is a stability boundary, which separates the stable flow region from the surge conditions. Typically, the safe operation is ensured by setting a surge avoidance line, which prevents operation near the actual surge line. If the blowers operating point is at the surge avoidance line, blower or external safety system will take actions to force the operating point back to the safe operating region. As the actual surge line may not be that well known, it is necessary to provide large enough margin between the avoidance and actual surge line, to make sure that the blower will not go into a surge due to disturbances in the system. This strategy is called surge avoidance and it limits the potential operating range and the performance of the blower in the low flow region. (Gravdahl & Egeland 1999)

The excess limitation of the blowers operating range due to surge margin can be mitigated by using surge detection. The incipient surge can be detected by monitoring variations in the inlet or outlet conditions and comparing them to the threshold values that define the incipient surge. The control actions are taken when the incipient surge is detected, rather than avoiding the surge condition by limiting the operation to a safe distance from the surge line. This method of surge detection and avoidance has some disadvantages as its safe operation requires large control forces and fast-acting control system for ensuring that the blower will not go into a full surge cycle. In addition, the behavior during the incipient surge is not similar between different blower designs, so the applicability of surge detection and avoidance is blower specific. (Gravdahl & Egeland 1999)

Third option for dealing with surge is the use of active surge control. As oppose to avoiding the surge condition, active control attempts to stabilize the unstable flow conditions, by commonly using throttle or close coupled valve as an actuator (Yoon et al. 2010). Gravdahl et al. (2002) showed that the operation in the unstable region can in theory be stabilized by using the variable frequency drive as an actuator for active surge control of centrifugal compressors. In practice, the motor-drive system will set physical limitations for the implementation of this type of control. These limits include maximum speed and torque of the drive as well as the maximum ramp times for speed and torque. Another challenge is that the control system requires fast feedback from the mass flow measurement. Simulations indicated that the time delay of the online mass flow measurement should be at maximum 50ms. Additional issues include limitations in the power supply, vibrations and the drive power loss. (Gravdahl et al. 2002)

During the startup of a centrifugal blower, the surge needs to be accounted for as the blower will pass through the surge region as it accelerates to its operating speed. Multistage blowers that use inlet throttling as a blower control, are sometimes started with the valve completely closed during the acceleration to the operating speed. After the speed is reached, the valve is opened, and the blower will pass through the surge. Another method for inlet throttled multistage blowers is to keep the valve at its minimum opening to mitigate the pulsation that occurs as the blower passes through the surge. Alternatively, the centrifugal blowers equipped with a blow-off valve can startup against low backpressure by keeping the blow-off valve opened during the acceleration. Blow-off valve can be also used for protecting the blower from surge, if the surge avoidance or control systems fail. (Jenkins 2013)

3.2 Blower control

For multistage centrifugal blowers, the blower output is typically controlled by throttling the air flow at the blower's inlet with a control valve. In typical operation of parallel multistage units, most blowers are operated near their BEP, while one of the blowers is controlled with inlet throttling, to adjust the air flow rate to the process. This is because there will be a significant drop in multistage units' efficiency while being operated outside its optimal operating range and the fact that inlet throttling is about creating additional pressure drop at the blower's inlet, which further reduces the blower's efficiency. (Rohrbacher et al. 2010)

More efficient way for modulating centrifugal blowers' output is to use inlet and outlet guide vanes instead of control valves. The guide vanes at the blower's inlet pre-rotate the air flow before it enters the blower's impeller, reducing the blower's output. (Spellman 2013) Apart from changing the air flow entering the blower's impeller, centrifugal blowers can also be controlled from the outlet side. The variable diffuser vanes control the blowers air flow rate by altering the conversion of kinetic energy into the static pressure. Varying the angles of the diffuser vanes moves the blower's characteristic curve horizontally. (Jenkins 2013) Using both of these methods together enables operation at blower's highest efficiency over a wider range of flow rates (Spellman 2013).

In addition to restricting or altering the air flow at the blower's inlet or outlet, adjusting the blowers rotational speed can be used for controlling the blower's output. Amerlinck et al. (2016), used the affinity laws for the calculation of the centrifugal blower curves with varying rotational speeds. Due to compressibility of air, the affinity laws do not necessary hold true for centrifugal blowers. Jenkins (2013) states that by applying affinity laws, reasonable accuracy can be achieved and for a sake of simplicity, affinity laws are assumed to hold true in this section of the study. The affinity laws are presented as follows:

$$\frac{N_1}{N_2} = \frac{Q_1}{Q_2}, \qquad \frac{N_1}{N_2} = \sqrt{\frac{p_1}{p_2}}, \qquad \frac{N_1}{N_2} = \sqrt[3]{\frac{P_1}{P_2}}$$

In which, the N_i is the rotational speed, Q_i is the air flow rate, pi is the discharge pressure and the P_i is the power drawn. The subscript 1 refers to the initial state and the subscript 2 to the new condition. By applying these laws, set of blower curves can be derived from a single constant speed line. These laws show that there is a cubic relation between the blower speed and the power drawn. This is due to variable torque nature of centrifugal blowers, which means that there is a square-law relation between the torque and the speed. The variable speed operation often offers energy-savings potential for variable torque application, as theoretically relatively small reductions in rotational speed yield large energy savings. For example, 10% reduction in blowers speed would reduce the power drawn by 27%, assuming that the affinity laws hold. (Spitzer 2012) The variable speed control is typically achieved by using VFD for controlling the blower motor. Motors and VFDs are further discussed in the following chapters.

3.3 Motors and variable frequency drives

3.3.1 Electric motors

According to Jenkins (2013), around 90 % of WWTPs total energy is consumed by electric motors. As the blowers, or to be more precise, the motors running the blowers are the largest single user of this electricity, their optimal sizing and operation is vital for improving the energy efficiency of the treatment process. The operational characteristics of the electric motors vary depending on their type and design. The selection of the motor is dependent on the torque and speed requirements of the blower as well as the operational patterns and ambient conditions. Additional factors worth considering are the motors efficiency and power factor. For the system design, correct motor selection and sizing according to motor characteristics is important as it directly affects the systems energy consumption and control capabilities. (Jenkins 2013) AC motors are typically categorized into two main groups based on their operating principle: induction and synchronous motors (Giri 2013) An example of the structure of synchronous permanent magnet (PM) motor and induction motor are shown in the figure 20.

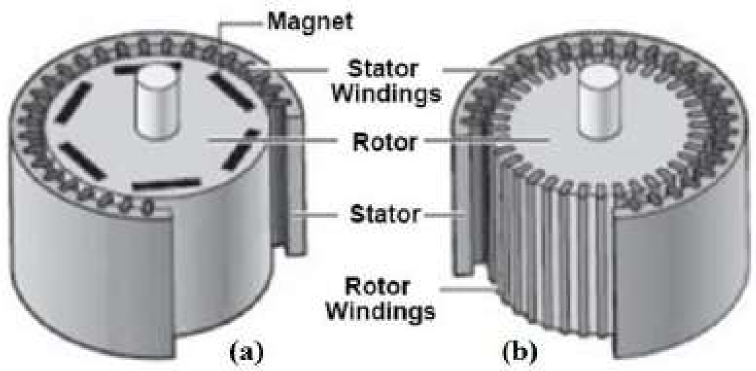

Figure 20. An example of basic structure of PM synchronous (a) and induction motor (b) (Al-Ibraheemi 2018).

In both induction and synchronous motors, the motors stator generates a rotating magnetic field as it is supplied with a polyphase AC voltage. This field will rotate at a synchronous speed, which depends on the supply frequency and the number of pole pairs in the field:

$$N_s = \frac{120 * f}{n_p}$$

Where N_s is synchronous speed in rpm, f is the supply frequency in Hz and the n_p is the pole number. Example of rotating stator field of a three-phase two pole motor is shown in the figure 21. In induction motors, this rotating field will induce torque-producing currents in the rotor as long as there is relative motion between the rotor and the rotating field. If the rotor would be rotating at a synchronous speed, no currents would be induced, and therefore no torque would be produced. This is why an induction motor will always run at a speed which is slightly less than the synchronous speed. This relative speed difference is quantified by slip:

$$s = \frac{N_s - N_r}{N_s}$$

Where s is the slip, N_s the synchronous speed and the N_r the rotor speed. This is important factor as the torque-producing currents are proportional to the rate at which the rotor conductors are cut by the rotating magnetic field and therefore they are proportional to the slip. If the motor is operated at a steady speed and the load is increased, the rotor will slow down and the slip will increase. Consequently, the rotor current and torque will increase to a point which the produced torque equals the load torque. Induction motors are designed so that they produce their full-load torque with a small slip value. While operating a motor with its nominal load, the rotor current will be at its safe maximum continuous value and increasing the load and consequently the slip, would start to overheat the rotor. While the induction motors are inherently self-starting, it is important to consider what is happening during the startup from a standstill. As the rotor speed is zero during the startup, the slip and the rotor current will be at their maximum. As the rotor starts to pick up speed, the slip and the current will decrease until the rotor reaches steady speed. (Hughes & Drury 2013) These high inrush currents can be quite significant, and they will subject the motor to a high thermal stress and can cause large voltage drops on the power supply.

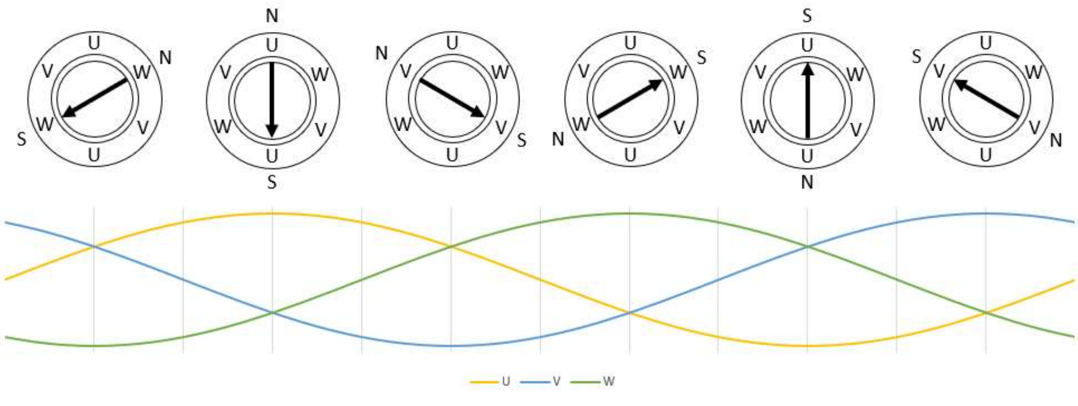

Figure 21. Rotating stator field of a three-phase two pole AC motor.

For synchronous motors, the operating principle is a bit different from an induction motor. Instead of the currents being induced in the rotor, the rotor will have its own magnetic field generated by a DC voltage or by permanent magnets. These will cause the rotor to lock on with the stators rotating field and the rotor will be always rotating at the synchronous speed of the stator field. The operating speed of synchronous motor will remain constant with varying loads, although the maximum torque will be limited. The maximum torque is limited to a point past which the rotor will be forced out of synchronism with the rotating field and the motor will stop. For producing a steady torque, rotor of a synchronous machine must be precisely at the same speed as the rotating field. This means that the synchronous motors are not inherently self-starting, as during the startup from a standstill, the rotor will not be able to synchronize with the rotating field, due to rotor and stator field passing each other and creating pulsating torque with an average value of zero. For most synchronous motors, the startup is implemented by using a rotor cage, which allows the motor to startup similarly to an induction motor, but as the rotor gains enough speed for synchronization, no current will be induced to the cage as the slip will be zero. (Hughes & Drury 2013)

For both induction and synchronous motor, the difference in the rotor speed and the synchronous speed defines the mode of operation of the motor. If the slip is between 0 and 1, the induction motor will be in motoring and the torque produced will be in a direction of rotation. For synchronous motor the slip will be zero during the normal operation. If the rotor speed exceeds the synchronous speed and the slip becomes negative, the motor will be acting as a generator, feeding the energy back to the supply. The negative slip can be obtained by either rotating the motor shaft faster that the synchronous speed, or by reducing the supply frequency and thus the synchronous speed. The negative slip will cause a braking torque, which should be kept in mind while decelerating the motor. (Moorthi 2005)

Jevremović & Jeftenić (2005) categorize different braking methods of electric motors, into three categories: inertial, soft and active braking. Inertial braking is about letting the motor to coast to stop, so the deceleration depends entirely on the connected load and mechanical losses in the motor. With soft braking, the motor speed is reduced gradually by giving linearly falling speed reference to the motor controller. With this method, it is not possible to dissipate the braking energy completely at the motor, and therefore the maximum braking torque cannot be obtained. In contrary, the third option of active braking can provide the maximum braking torque as it dissipates maximum power in the stator windings. This requires that the stator currents are at their maximum, which can be limited by the capabilities of the VFD controlling the motor. (Jevremović & Jeftenić 2005)

All electric motors are subject to thermal constraints and the maximum thermal conditions can be reached through different operating patterns. The thermal time constant is an important parameter, as it describes the rate of temperature rise. For large motors, the thermal time constant can be more than an hour and for medium-power motors thermal time constant is in the range of tens of minutes. When selecting a motor for a particular application the required power needs to be estimated. The general procedure for application where the speed remains constant and load varies, is to assume that the thermal losses vary with the square of the load and the required motor size can be estimated from the root mean square of the power cycle. In addition to this estimation, the motor must be able to provide the maximum torque required in the overload conditions. The motor rating is typically provided as continuous rating and short time rating, the continuous being the maximum load for an unlimited time and the short time rating being typically the maximum loading for either 10, 30 or 60 minutes starting from ambient temperature. The fact that the rating is given for a specific temperature condition is important as if the motor has been overloaded previously and it has not cooled down, the overloading will be limited. Manufacturers typically also give motor a duty class, which can be based on one of the eight standard duty cycles defined in the IEC 60034-1. Broadly, the duty can be defined as a continuous, intermitted or special duty. In applications where motor speed is controlled with a VFD, the duty cycle can be very important as the thermal time constants related to power electronics are much shorter than for the motors. (Hughes & Drury 2013)

3.3.2 Variable frequency drives

The task of the VFD is to convert the constant power supply into the voltage and frequency, that are required for achieving the desired mechanical output from the motor. The desired output can be either speed, torque, position of the motor shaft or other system variable that is affected by the motors mechanical output. (Hughes & Drury 2013) Typical structure of a VFD is shown in the figure 22. A VFD operating with an AC power supply, consists of two power conversion processes. The first conversion takes place in VFDs rectifier which converts the grid AC into DC voltage and supplies it through the DC link to the inverter where the DC is converted back into synthetic AC voltage. The DC link between the two power converters acts as an energy storage and it smooths the DC voltage produced in the rectifier. (Giri 2013) In the inverter side, each output phase is connected to the DC link by two semiconductors, which by alternating on and off states, connect the phase to either +DC or -DC voltage. By adjusting the timing of these switches, inverter can output different voltage waveforms to the motor. It is important to note that the output will not be a pure sine wave, but only an approximation, which's quality depends on the inverter design and control strategy used for generating the desired waveform. The on and off nature of the inverters process results in waveforms with short rise times and multiple harmonics that may require additional smoothing with output filters. (Spitzer 2012)

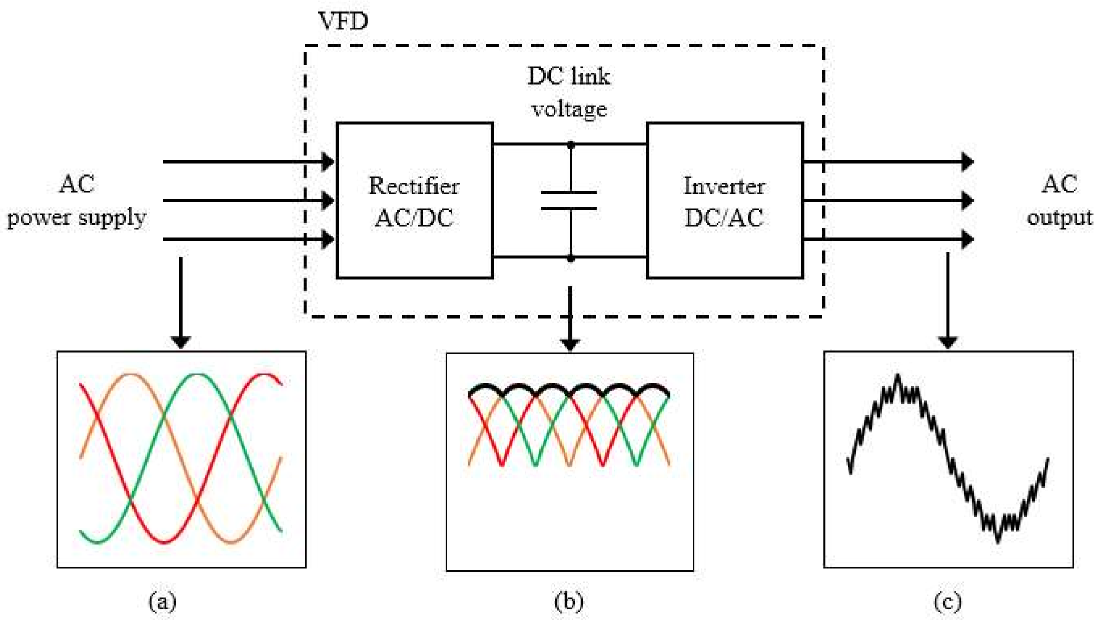

Figure 22. Typical structure of VFD and depiction of VFDs power conversion process. Three phase AC power (a) is converted into DC (b) in rectifier and supplied through DC link into inverter which outputs desired waveform (c).

The switching frequency is the rate at which the inverters semiconductors operate. It is closely related to the quality of the output waveform and the losses in the inverter and the motor. Higher switching frequencies improve the waveform and consequently decrease the losses at the load side, but on the other hand, the related switching losses cause thermal stress on the inverter. This means that there generally is a tradeoff between the inverter losses and the output waveforms quality. (Onederra et al. 2017)

For controlling the motor, VFDs use different control algorithms, which can be broadly categorized into simple scalar controls and more sophisticated vector-based controls. The main difference between these two larger groups is that the scalar control is based on relationships, that are only valid in a steady state, whereas relations used in vector controls are also valid under dynamic conditions. This means that as the scalar control only controls the magnitude and frequency of the voltage, current and flux linkage vectors, the vector-based algorithms control also the instantaneous positions of these space vectors. (Giuseppe et al. 2004) The scalar controls are more suitable for applications, which do not require high dynamic performance of speed and torque control whereas the vector controls are preferred in more demanding applications that require the precise motor control under dynamic conditions. The drawback on the traditional vector controls is that they require feedback from the motor. Alternative to the scalar and vector controls is the Direct Torque Control, which controls the motors magnetizing flux and torque directly, without the need for a speed feedback from the motor. (ABB 2011)

The conversion process in the rectifier generates harmonics on the line side of the VFD. The harmonics can be an issue, as the distortion may have negative effects on the equipment connected to the same supply. The limits for the harmonic distortion at the point of common coupling, are defined in the standard IEEE-519. In the wastewater industry, it is common to use the IEEE-519 limits for each VFD's termination point, rather than just for the point of common coupling, which typically would be the plants supply transformer. There are several methods for reducing the line side harmonics, such as line reactors and VFDs with active front ends that dynamically cancel the harmonics. (Jenkins 2013)

The quality of the VFDs output waveform is important, as harmonics and voltage spikes can deteriorate the efficiency and potentially damage the motor (Spitzer 2012). The negative effects of the generated waveform and disturbances caused by the inverter can be mitigated by using output filters. Although the output filters offer many benefits such as increased efficiency and lifespan of the motor, they oppose some challenges for the motor control strategy. This is because as there is a filter in between the VFD and the motor, the voltage and current measured at the inverter circuit will no longer be accurate for the motor control. The difference between the values at the filters input and output needs to be accounted for by including the filter in the VFDs control and estimation algorithms. (Guziński et al. 2015)

3.3.3 Blower application considerations

For the blower application, the requirements for the motor are set by the required speed, torque-speed characteristics, inertia of the load and the control type of the blower. The conventional blower technologies typically use standard motors, which have nominal speeds ranging from 1000 to 3600 rpm. Conventional blowers can also use belt or gear drive for increasing the speed. If the blower is controlled with a VFD, the motor can be operated with a higher speed than nominal to some extent, if the motors rating allows it. For high-speed turbo blowers, special high-speed motors and VFDs are required for reaching the required speed range. (Jenkins 2013)

The torque-speed characteristics depend on the individual blower's design, but the general characterization can be done based on the blower type. Positive displacement blowers are constant torque loads and centrifugal blowers are variable torque loads. For a constant torque load, the torque will remain constant regardless of the speed. As the mechanical output is the product of torque and speed, there is a linear relation between the power and speed. Conversely for variable torque load, the torque will increase in a square of speed and thus the power will increase in cube of speed. If the motor is variable speed controlled and the motor is rated for a continuous operation at the nominal speed, and the actual operating point is at half speed, the motor loading will be significantly different depending on the load type. For a variable torque load, the motor will be operated at only 12.5% of its rated power, whereas for constant torque application the motor will be at 50% of its rated load. (Hughes & Drury 2013) Having the motor underloaded is not desirable, as the efficiencies of electric motors decrease significantly below 50% load, whereas motors power factor will be decreasing even sooner. This again comes down to properly sizing the motor, so that it will be operated within its good efficiency range, while still being able to provide sufficient torque during the peak loadings. (Spellman 2013)

Regarding the costs of operating the blowers, the motors' effect on plants power factor should be considered. Operation of induction motors delays the current waveform, causing a phase shift between the current and voltage waveforms. This phase shift causes less net power to be transmitted, leading into increased power requirement. The ratio between the active power that does the actual work and the apparent power which includes the reactive power is called the power factor, which basically describes how much of the power supplied is used for the actual work. If the plant has a poor power factor, which is often caused by the motors running below their rated load, extra fees may be paid for the electric utility. In addition to saving in electricity costs, improving low power factor is desirable, because it decreases the system losses, increases the capacity and improves the voltage. The power factor can be improved by resizing the motors or by adjusting the system, so that the motors could operate closer to their full load. This might be challenging, as the correct operation of the blowers is the primary concern. (Spellman 2013) The use of VFD for motor control can improve the motors power factor, but in turn it will generate harmonics and decrease the system efficiency at stable full-load operation (Hughes & Drury 2013). Because of the nature of the blower application in wastewater aeration, the decrease of efficiency at full load is rarely an issue, due to variation in the power requirements.

4 High-speed turbo blowers

After their introduction to the wastewater market in the early 2000s, the high-speed turbo blowers have proceeded from being an industry unknown to the preferred alternative at the treatment plants (Zahller & Koch 2012). High-speed turbo blower is typically provided as a package, where blower, high-speed motor, VFD, electrical filters and programmable logic controller (PLC) are within a single enclosure, which is generally air cooled. Isolation and blow-off valves are located outside the package, on the blower's discharge. (Bell, et al. 2014) Diagram of a typical high-speed turbo blower configuration is shown in the figure 23.

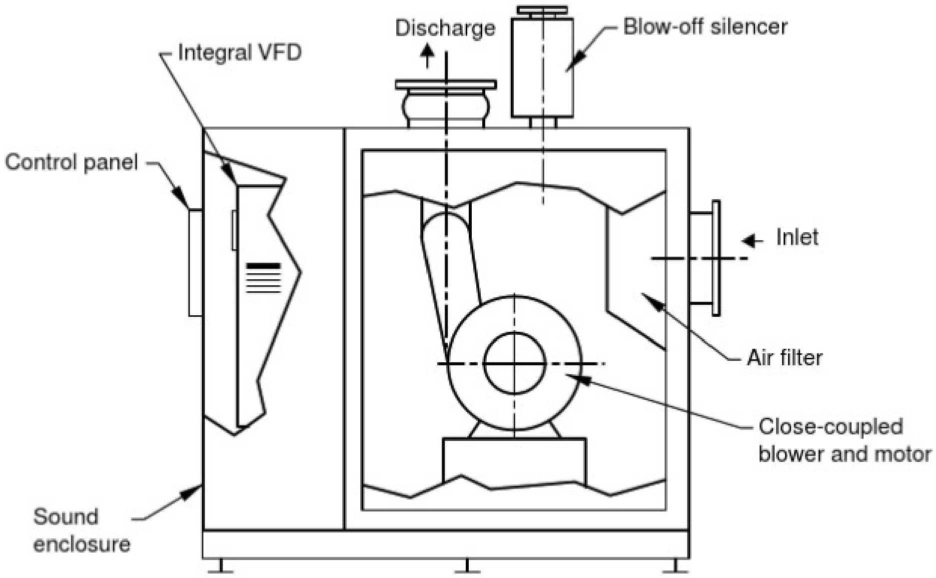

Figure 23. Typical high-speed turbo blower package (Jenkins 2013).

Case studies by Bell et al. (2014), Rohrbacher et al. (2010) and Schilling & Turriciano (2011) have shown the potential benefits of high-speed turbo blowers, but also highlighted some issues related to the new technology.

4.1 Characteristics

High-speed turbo blowers are direct drive single-stage centrifugal blowers, so the shape of their blower curves resemble the ones of the conventional centrifugal blowers. They are also subject to the same operational limits based on their impellers design, surge limiting the operation at low flow rates and choking restricting the flow production at high flow rates. The motors and VFDs used in high-speed turbo blowers are also subject to same thermal constraints as their standard alternatives, although their operation with high rotational speeds and frequencies adds some special considerations to the system design.

High-speed operation is beneficial, as the dynamic efficiencies of the blowers improve with the increasing speeds (Bell & Abel 2011). High-speed units are generally seen as having high efficiency over a wide range of discharge pressures and flow rates, typically having 10 to 20 percent better efficiency than multistage centrifugal or positive displacement blowers. Their turndown is up to 50% and although their efficiency should drop only a little over the operating range, not much documentation exists about their actual efficiency at turndown. In terms efficiency and turndown, high-speed units are on par with single stage geared units utilizing dual point control. Dual point controlled geared units are claimed to have a bit better turndown and their efficiencies are claimed to remain quite constant with the dual point control. (Spellman 2013)

Although the geared units that utilize dual point control might provide more energy efficient operation in some cases, their disadvantages are the higher initial cost, mechanical complexity and maintenance. Only maintenance for high-speed units is the air filter change, whereas geared units require oil/oil filter changes in addition to air filter maintenance. Due to large oil reservoir, the maintenance of the geared units can be relatively costly, depending on the frequency of the oil changes, which vary between installations from every three months to several years. The mechanical complexity of the geared units comes from the required pressure-lubrication oil system and the blower controls, which make them more intricate than other technologies. In contrast, the high-speed units have only a few moving parts, but they have significantly greater electrical complexity, than the conventional technologies. (Rohrbacher et al. 2010)

Major benefit of the high-speed units is their small footprint, which enables the reduction in required building footprints by at least 25% (Bell & Abel 2011). This reduces the building costs and might make building expansions unnecessary for the plants that are looking for increasing their aeration capacity. As the high-speed units are typically provided as an integrated package, their installation is relatively easy and their acoustic enclosure together with the high-speed technology provides quiet and vibration free operation. The cooling system is also a part of the package, so there is no need for the external cooling systems. (Rohrbacher et al. 2010) The cooling of the enclosure is still an important aspect for the manufacturers, as both motor and VFD generate heat and are subject to thermal limitations.

As already mentioned, all centrifugal blowers, including the high-speed units are subject to surge in low flow rates and they will pass through the surge region during the startup. Although Jenkins (2013) claims that the VFD controlled blowers can pass through the surge fast enough without damaging the blower, Zahller and Koch (2012) emphasize the importance of always starting a high-speed unit against no backpressure for preventing the premature failure of the unit. This leads to the need of assessing the piping configuration to ensure that units are started up against no backpressure. Starting a high-speed unit against full backpressure might lead into ultimate failure of the unit. (Zahller & Koch 2012)

Coasting to full stop might take a long time for a high-speed turbo blower. This is due to high-speed of the rotor and low losses in the bearings. Smirnov et al. 2016 performed rundown test for a high-speed machine operating with a nominal frequency of 250 Hz, which showed that only a fraction of the bearing losses resulted in the deceleration of the motor, leading into extremely long deceleration time. It should be noted that no load was connected to the motor and for motor operating a blower, the load would naturally help with the deceleration.

Under normal operation, high-speed turbo blowers do not experience large step changes in their loading, so fast speed or torque response is not necessary from the motor control. Also, there is no requirement for high starting torque in the high-speed units. (Halkosaari 2006) These aspects will simplify the motor control, but the optimal operation of high-speed motor system will still be quite challenging. The high-speed motors and VFDs will be discussed in the following sections.

4.2 High-speed technologies

4.2.1 High-speed motors

The limiting factors for the maximum output power of a high-speed machine depends on the operating speed. For the high-speed machines operating at relatively low speeds of less than 20 000 rpm, the power is limited by the maximum mechanical stress in the rotor and for the machines operating at speeds above 20 000 rpm, the power is mainly limited by the losses in the machine. Although different types of electric motors are used in high-speed applications, they all are subject to more or less the same mechanical and thermal constraints. (Moghaddam 2014) In high-speed turbo blowers, both induction and PM motors are used, although the PM motors are more prevalent due to their improved power factor and good efficiency. (Halkosaari 2006)

For an electric motor, the torque is interconnected to the machine's dimensions, and as the torque is proportional to the mechanical output and inversely proportional to the supply frequency, using high frequency current enables design of compact motors. During the high-speed operation, the motors rotor is subject to centrifugal forces, which cause mechanical stress that is proportional to the rotors circumferential speed and the rotor mass density. The circumferential speed naturally depends on the angular speed and the rotor diameter, which in turn is determined by the motor torque. If the rated rotational speed is increased, the rotor diameter needs to be decreased, which consequently means reducing the rated torque and power of the motor. Due to this the high-speed machines physical sizes and power ratings reduce with the increasing rotational speeds. For machines operating above 20 000 rpm, the limitation for the maximum output power comes from the thermal limits due to increased losses in the motor rather than from the mechanical stress on the rotor. These losses in the motor are influenced by the VFD supplying the power to the motor, so the quality of the VFDs output should be considered while designing a high-speed system. (Moghaddam 2014)

4.2.2 Friction-free bearings

With the rotational speeds ranging from 15 000 to 50 000 rpm, the standard roller bearings cannot provide sufficient bearing life, which is why friction-free bearings are necessary for the high-speed operation. Two different bearing technologies are utilized in these machines: magnetic and airfoil bearings. (Irving & Ibets 2013) The friction-free bearings contribute to the high-speed units' advantages over the conventional technologies, as they reduce the maintenance requirements by eliminating the need for the lubrication system, which can be relatively complex (Rohrbacher et al. 2010).

The operation of an airfoil bearing is based on the dynamic pressure created between the shaft and the bearing. During the startup, there will be a contact between the shaft and the bearing as there will be no pressure to levitate the shaft. The shaft loses contact with the bearing as it reaches the speed required for producing enough pressure, which is typically at around 2000 to 5000 rpm. For reducing the wear of the bearings during the startup, the contacting surfaces are coated in order to provide solid lubrication. Although the airfoil bearings can withstand many starts and stops, the coating will degrade overtime with repeating cycles of starting and stopping of the blower. (Irving & Ibets 2013) As one of the drawbacks of the high-speed turbo blowers is the limited available power range, the use of airfoil bearings may restrict the blower design, as their capability to support larger shafts is limited. Airfoil bearings are also seen as having reliability issues, while operating at the WWTPs demanding environments. High ambient temperature and the contact during starting and stopping of the blower are specific concerns for the plants and although the concerns might be unfounded, they have been limiting the adaptation of the technology. (Severson & Mohan 2017)

In Active Magnetic Bearing (AMB) systems, the motor shaft is supported by a magnetic field, which is generated by dynamically actuated electromagnets. The Magnetic Bearing Controller receives feedback of the shaft location and actively adjusts the power to each magnet for keeping the shaft at the center of the assembly. This type of system allows the automatic corrections to imbalances as well as possibility to monitor different performance characteristics as the feedback sensors are already a part of the AMBs. For example, the AMBs sensors can be used for estimating the proximity to surge. Additionally, the sensors provide important data that can be used for monitoring the overall health of the equipment. As the AMBs require power to operate, a fail-safe mechanism in case of power outage is required. Typical solution is to use uninterruptible power supply (UPS) for ensuring the operation of the bearings, until the motor is stopped. Alternative to UPS is the dynamic braking of the motor with the VFD and supplying the generated power to the bearings. For both of these methods, back-up bearing can be used. The back-up bearing is a roller bearing, on which the shaft will land on, if the UPS or the VFD fails during the power outage. (Irving & Ibets 2013)

4.2.3 High-speed VFD

Beer et al. (2006) defines the following requirements for the VFDs output: independently variable voltage and frequency, good phase balance between the voltage and current with minimal DC component, limited harmonic content, low common-mode voltage and low dV/dt into the motor. These requirements are to minimize the motor losses and stress on the motor insulation but abiding to these requirements can be challenging while operating with high output frequencies. It does not help that the high-speed motors are more sensitive to the supply power quality than the standard motors (Moghaddam 2014).

The required output frequencies from the VFD can be quite high, while the switching frequency might be limited by the thermal capabilities of the inverter. The ratio between the switching frequency and output frequency can be used for characterizing the control system stability, low ratios of below 10 requiring higher frequency response from the VFD. High output frequencies together with insufficient switching frequency will increase the harmonic distortion and the losses in the motor, due to low number of switching operations per fundamental period. In other words, the quality of the waveform decreases with lower switching frequency to output frequency ratios, and the significance of optimal switching frequency is pronounced in high-speed applications. This is why many high-speed systems require output filter between the VFD and the motor, for reducing the associated current and voltage harmonics. (Walz et al. 2019) The modulation strategy should also be considered as it along with the switching frequency, which might be limited by the hardware capabilities, determines the quality of the output waveform. The objective for optimizing the modulation would be to produce waveforms with least harmonics, with the minimum switching frequency for minimizing the losses in the VFD. (Halkosaari 2006)

In high-speed applications, getting feedback from the motor can be challenging due to limited capabilities of speed sensors and encoders. Because of this, VFDs have to rely on open loop controls in high-speed applications. Due to characteristics of the blower application, open loop controls are sufficient, but because of the high rotational speed, the control is not as simple as it is for the standard motor. (Halkosaari 2006) The simple U/f control is shown to work with the high-speed motors, but as it does not control the stator currents, the motor performance is not optimal. The lack of control of the stator currents can cause distortion of the stator flux linkage resulting in torque ripple and decreased static and dynamic accuracy. (Walz et al. 2019) While blowers are often operated under partial loads, some optimization of the motor voltage, current and losses can be done through the VFDs control algorithms. One method is to control the motors power factor to 1 by minimizing the reactive current. This method indeed decreases the current, but it usually does not find the optimum. Another method is to use voltage iteration, where small changes to motor voltage are made to find the minimum motor current. These two methods have the advantage of being independent from the motor parameters. Third alternative which in turn requires the knowledge of the motor parameters, is to calculate the optimal voltage based on the motors steady state equations. For this method, the effects of the commonly used sine-filter can be included in the optimization. (Halkosaari 2006)

4.3 Special considerations

For high-speed turbo blowers that utilize airfoil bearings, the number of daily starts and stops should be minimized, because of the wear of the coating on the contacting surfaces. The airfoil bearings should last around 20 000 starts and if it is estimated that the blower would require over six start/stop cycles per day, either idle mode or alternative blower technology should be considered. (Bell et al. 2014)

It is vital that the high-speed turbo blower starts against no backpressure and the blow-off valve must be included as a part of the configuration. The startup against backpressure, will lead to vibrations and may prevent the blower from ramping up to its operating speed. (Zahller & Koch 2012) Blow-off valve can also be used for surge protection as opening the valve when impending surge is detected reduces the pressure and consequently increases the air flow rate (Jenkins 2013).

High-speed turbo blowers are often installed as retrofits to existing aeration systems and it should be noted that the existing electrical system may have not included VFD previously. If this is the case, including a harmonic filter as a part of blower package should be considered. For new installations, the harmonic filter may be included at the point of common coupling, so there should be no need for additional filter. (Zahller & Koch 2012) Onsite harmonic test should be carried out to confirm that distortion levels are within the limits set by the applicable standards. Also, the power reliability at the WWTP should be evaluated, as the high-speed VFDs are sensitive to fluctuations in the supply voltage. (Bell et al. 2014)

If a high-speed turbo blower is retrofitted to a system, with the intention of it operating together with conventional blowers utilizing different control methods, the control system might require some modifications. The VFD controlled turbo blower will behave differently than for example inlet throttled multistage unit, which might lead to complex interactions between the blowers as they try to control the same variable, with different response times and control loops. Solution for this might be to operate the existing blowers at a constant speed as a base load while the turbo blowers would adjust their airflow rate according to process requirements. (Zahller & Koch 2012) Although this would solve the problem with conflicting control strategies, the benefits of this implementation are case specific. The proposed solution of operating only one blower in modulating duty is also one of the critic concepts of the compressed air systems best practices (Hall 2018).

As the high-speed units have motors and VFDs that are very specific for particular blower, the premature failure of either of these can results in significant costs as the replacement are not available “off the shelf” (Rohrbacher 2010). For the plant operators, it should be noted that the high-speed turbo blowers are a more hands-off equipment than traditional technologies and the plants staff might be unable to perform repairs for the blower. Typically, the repairs are done by the blower manufacturer and in some cases, the spare parts need to be shipped overseas. (Bell et al. 2014) This emphasizes the need for reliable operation of the high-speed VFD and motor.

5 Model development for aeration system and high-speed turbo blowers

5.1 Hermanninsaari WWTP

Hermanninsaari WWTP is located in the Southern Finland and it receives wastewaters from the nearby City of Porvoo and local small industries. The amount of treated wastewater varies between 4 000 000 and 5 000 000 m^3/a. In the plants environmental permit from 2015, it is estimated that the influent loading would increase 21% from 2011 to 2030. This would surpass the plants design loading. (AVI 2015) The current design parameters for plant and aeration system are presented in the table 2.

Table 2. Design parameters of the Hermanninsaari WWTP [Adapted from: AVI (2015) & Renkonen (2014)].

Average influent flow rate	13 200 m^3/d
Maximum influent flow rate	37 000 m^3/d
Design population equivalent	38 600
Aeration air flow rate at design point	4600 Nm^3/h
Pre-aeration air flow requirement	400 Nm^3/h
Pressure setpoint	91 kPa

The biological treatment process at Hermanninsaari WWTP consists of two identical treatment lines operating in parallel, each line being divided into 8 zones. Simplified schematic of the aeration system is shown in the figure 24. The zones 4-7 have diffuser grids and the aeration to each zone is controlled with butterfly valves. The piping branches are connected to a common header, in which the air is supplied with three high-speed turbo blowers. The aeration control system can be characterized as ammonia-feedforward control and the control of the blowers is based on a constant header pressure.

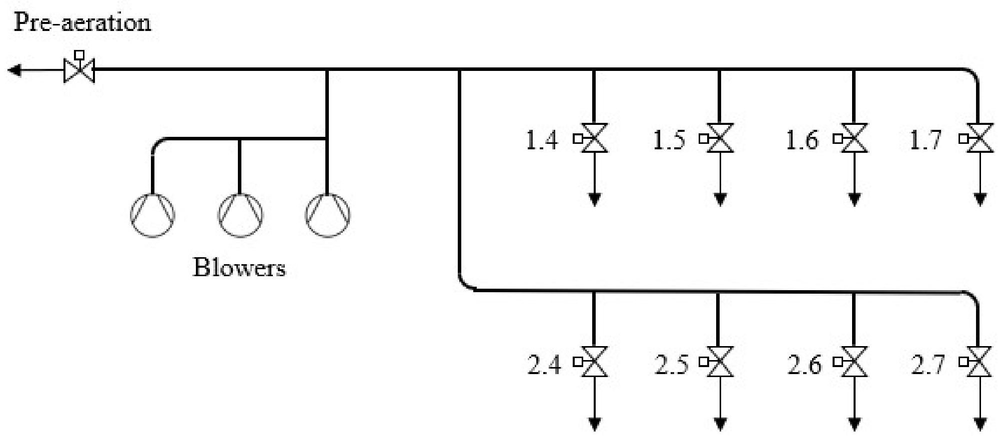

Figure 24. Simplified drawing of the Hermanninsaari WWTPs aeration system.

Both aerated volume and aeration intensity is controlled according to the NH_4^+-N measurement taken from the 5th zone of the second treatment line. The NH_4^+-N measurement determines if the zones 4 and 5 are aerated and sets the DO setpoint for the zones 6 and 7. For the 4th zone, the aeration starts when the NH_4^+-N reading exceeds 8 mg/l and stops when the reading drops below 6 mg/l. For 5th zone, the aeration starts at 5 mg/l and stops at 4 mg/l. The valves of zones 4 and 5 have fixed setpoints for their openings, as there is no DO feedback for the valve control. There is also a 15-minute delay in the starting and stopping of the aeration of the zones 4 and 5. For zones 6 and 7, the DO setpoint is determined according to the NH_4^+-N concentration, the setpoint being limited to be 1.5 mg/l at minimum and 3 mg/l at maximum. The valves of these zones will adjust their position based on the zone's deviation from the DO setpoint. During the startup of the secondary blower, the control valves remain at constant positions for 30 minutes. This is set in the blower systems control parameters, as a DO control interlock time, which is for ensuring stable operation during the change from single to two blower operation. In the control system, it is possible to distribute the DO setpoint between basins to adjust the oxygen profile of the aerobic section of the treatment line.

The three high-speed turbo blowers are controlled to keep the discharge pressure constant, the setpoint for pressure being 91 kPa. The blowers are operated so that one of the blowers serves as the primary blower that is operated constantly, one is a secondary blower, which is started when the primary blower reaches 97% of its nominal load and the third blower remains at standby. The duties are switched every week, so that the primary blower goes into standby, secondary becomes the new primary unit and the standby becomes the secondary unit. This means that each blower is operated for two weeks before resting for one week, the first week being intermitted operation as secondary blower and the second week continuous operation as a primary blower and the third week resting in standby. The control strategy of the primary and secondary blower is explained in more detail in section 5.3.2.

5.2 Collected data

The plant collects data on several process parameters. From the aeration process data is collected on pH, NH_4^+-N, DO, control valve positions etc. Historical data is available in hourly and daily scales, but the raw data is available on a minute scale for a limited time period. For this study, the raw data was used, which by the time of data acquisition was available for the time period of 4.12.2018 to 8.5.2019. The data for this period included the DO in the constantly aerated zones, NH_4^+-N, control valve positions and air flow rates within the aeration process. The data of the blower loadings was available for a shorter time period from 4.4.2019 to 20.5.2019.

The whole available dataset for valve positions was used for this study, although some minor filtering and scaling was required. The raw dataset of the valve positions, NH_4^+-N and DO measurements are presented in the appendix 1. The dataset includes periods of both high and low loadings and although the first half of the dataset is quite different from the latter part of the data in terms of NH_4^+-N loading, the whole dataset was used. This was done because based on the DO measurement and valve positions, the process was working as it was intended. It should be noted that the control valve of zone 5 in treatment line 1, was not in operation during the time period presented in the data.

5.3 Blower model

5.3.1 Modelling of the high-speed turbo blower

The purpose of the blower model is to approximate the relationships between different operating parameters, such as discharge pressure, air flow rate and blower speed. Different methods exist for modelling the blower characteristics and generally the model can be either physical-based or a numerical approximation based on the measurement data. For physical-based models, detailed knowledge on the particular blower's design parameters is required and although the resulting model might capture the operating characteristics quite well, the required data might be difficult to acquire. For numerical approximation, polynomial fitting to the manufacturers data is quite common method for finding the relationship between discharge pressure and flow rate. The additional blower curves with varying rotational speeds can then be generated from this single polynomial by applying different scaling methods. This is quite simple and efficient method for constructing the model and the required data is generally available from the manufacturers. An alternative to these two approaches is the table-lookup method, in which the additional blower curves are generated in between known blower curves by using linear interpolation. This method can yield more accurate results than the polynomial approximation, but it can be less efficient. For this study, the polynomial approximation was used. The table-lookup method was attempted, but it was deemed too slow for the longer simulations. The blower maps containing multiple blower curves in different speeds and curves presenting the relation between power, air flow rate and rotational speed were provided by the manufacturer.

Typically, aeration system models use affinity laws to describe the effects of varying blower speeds to centrifugal blowers' characteristic curve. This approach is suggested for example by Amerlinck et al. (2016) and Amaral, et al. (2017), but as for this study capturing the high-speed turbo blowers actual operating characteristics was important, an alternative approach was needed. Third order polynomial was fitted to data points of a single blower curve and additional speed lines were generated by using adjusted affinity laws. The adjustments were made to the relative rotational speed by scaling it to a range and using scaling factor to adjust the generated speed lines so that they would somewhat fit the manufacturer data. This was done because as the affinity laws by themselves do not take gas compressibility into account, the accuracy of the generated blower curves would deteriorate as the speed would diverge from the original curve. This method of scaling is a feasible solution, when modelling a specific blower type, but using the affinity laws may offer a good generalization, when the performance of the blower itself is not a vital part of the study. The resulting curves and the data points used to estimate the blower characteristics are shown in the figure 25. Similar method of polynomial approximation was used to relate the blower power to the air flow rate and rotational speed and the resulting curves with the used data points are shown in the figure 26.

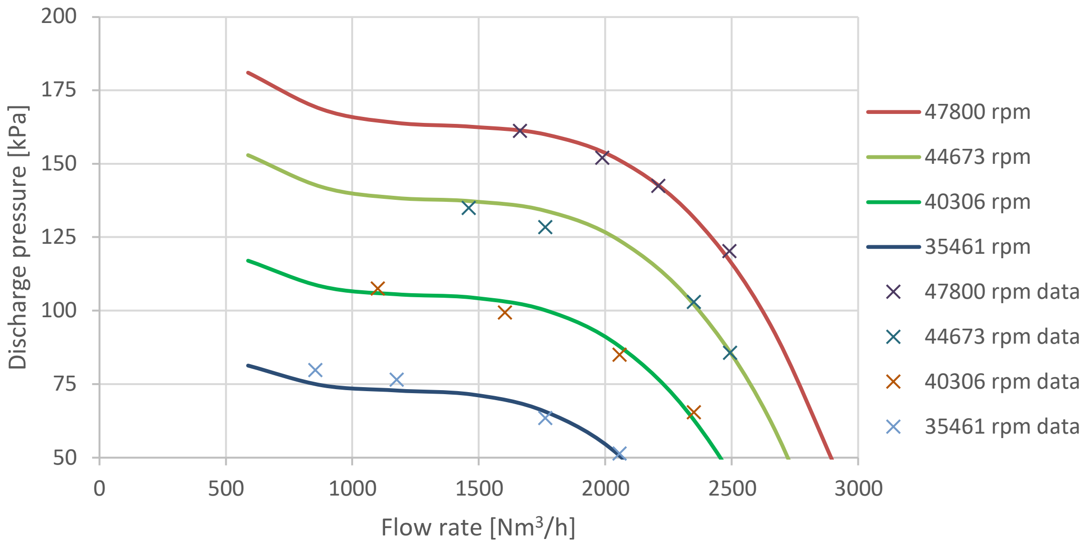

Figure 25. Polynomial approximation and reference data points of pressure vs. air flow rate with various rotational speeds.

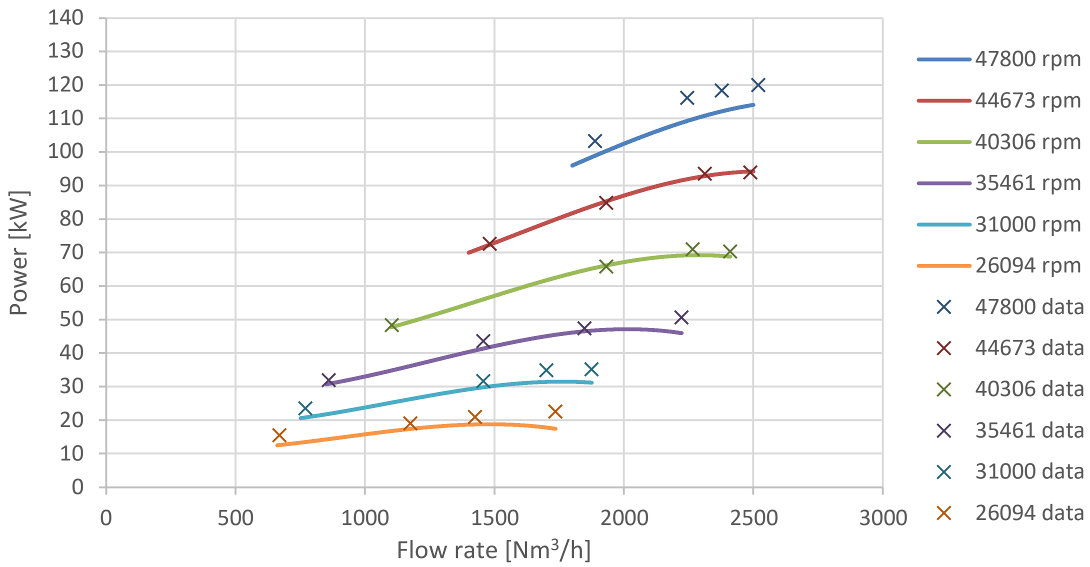

Figure 26. Polynomial approximation and reference data points of power vs. air flow rate curves with various rotational speeds.

The blower model is implemented in Simulink as a subsystem that takes rotational speed and the discharge pressure as inputs and solves the corresponding air flow rate and power, based on the polynomial approximations of the curves and the scaling of the relative rotational speed. The subsystem basically consists of one while-loop, which substitutes air flow rate together with the scaled rotational speed into the equations for pressure and power. Loop then incrementally increases the air flow rate, until the correct pressure is found. With this model, the blower controller can adjust the rotational speed for controlling the air flow rate, power or the system pressure.

It should be noted that the developed blower model assumes that the conditions at the blower inlet remain constant. The selected parameters for inlet conditions are: 15 ̊C inlet air temperature, with 0% relative humidity and 101.325 kPa inlet pressure. In physical based models, variations in inlet conditions would inherently be taken into account, but the model would still require input data of the inlet conditions, which is not readily available. If the data on the inlet conditions would be available, they could be included in the numerical model as different scaling factors, which would further increase the complexity of the model.

5.3.2 Modelling of the blower controls

As already mentioned, the aeration system in Hermanninsaari WWTP has three identical blowers that cycle between different duties. For the model, the system was simplified so that there are only two blowers from which one remains as the primary blower and other remains as the secondary blower. The cycle time between different duties is considered while analyzing the results. The blowers receive feedback signal of the header pressure and adjust their speeds for maintaining the pressure at its setpoint. The pressure feedback is also used for calculating the limit values for speed, as the power and surge limit will be changing based on the output pressure condition.

Although, the plant data indicates that there is a slight offset between the loadings of the primary and secondary blower, this is not considered, as it is likely related to the scaling of the data. The raw data scales the loading to a range of 50-100%, 50% indicating that the blower is not running. Because some of the blowers remain over 50% even at standby and some cap below 100%, it leads to believe that the offset is because of these small errors in the data. Also, because the blowers are pressure controlled, it is assumed that the load between the blowers is perfectly balanced.

For determining the start and stop signals for the secondary blower, separate subsystem is constructed. Initially the subsystem uses the following rules for determining the start and stop signal:

- if the loading of the primary blower remains over 97% for over 10 minutes, secondary blower is started

- once the secondary blower is started, it will be running for at least 30 minutes before it can be stopped, because of the DO control interlock, which prevents the valves from closing down.

- if both blowers are running and the loading of a single blower drops below 56%, secondary blower will be stopped

During the calibration, these control rules were slightly adjusted, as the correct start and stop cycles are important for achieving adequate model performance. Example of a start and stop cycle of the blowers is shown in the figure 27. and the model of the blower system is shown in the figure 28.

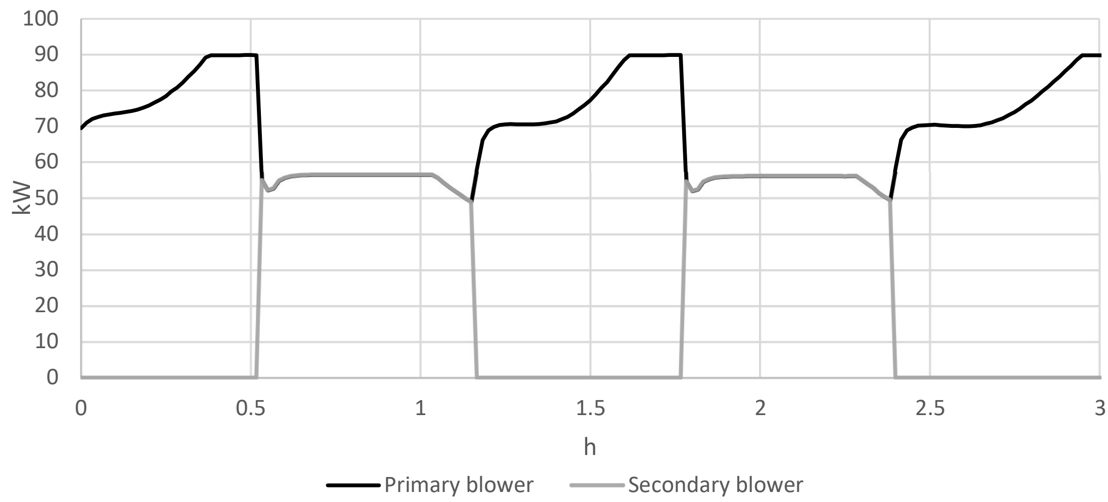

Figure 27. Example of start and stop cycles of the primary and secondary blowers.

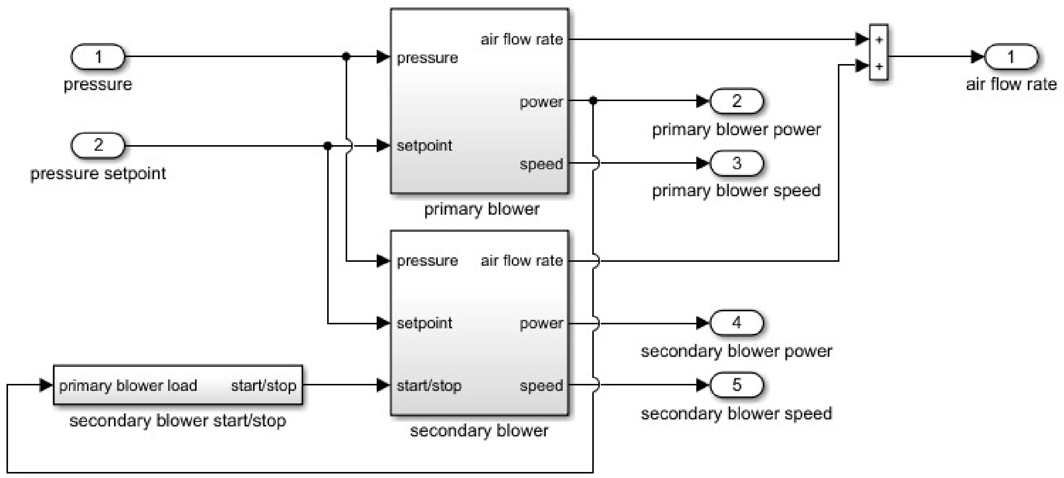

Figure 28. The model of the blower system.

5.4 Aeration distribution system model

The model of aeration distribution system is based on the model presented in the study by Harja et al. (2016). The pressure calculation is based on the difference in the air flow rate in the input and the output of the pressure system. The pressure is described with the differential equation:

$$\frac{dp(t)}{dt} = \frac{p(t)}{V}\left(Q_{in}(t) - Q_{out}(t)\right)$$

Where p is the pressure, V is the volume of the pipeline, Qin is the air flow into the system from the blowers and the Qout is the air flow rate at the pipelines discharge, described by the equation:

$$Q_{out}(t) = \sum_{i=1}^{9} k_i(t)\sqrt{p(t) * (p(t) - p_{stat})}$$

Where the p_{stat} is the static pressure at the aeration systems discharge and the k_i is the factor describing the pneumatic resistance of a specific piping section denoted with the subscript i. In the Hermanninsaari WWTP, there are 9 different piping sections branching out from the main header, each separated with their own control valves. In the model by Harja et al. (2016) the control valve position itself acts as a factor multiplying the k_i of a controlled pipe section. In this study, the parallel valves are modelled as a large single valve, yielding one value for the k_i. It would have been possible to give k_i values for each of the piping sections, but calibration of the model would have been challenging as it would have required data from each piping sections air flow rate, which was not available at the time. The model of the aeration distribution system is shown in the figure 29.

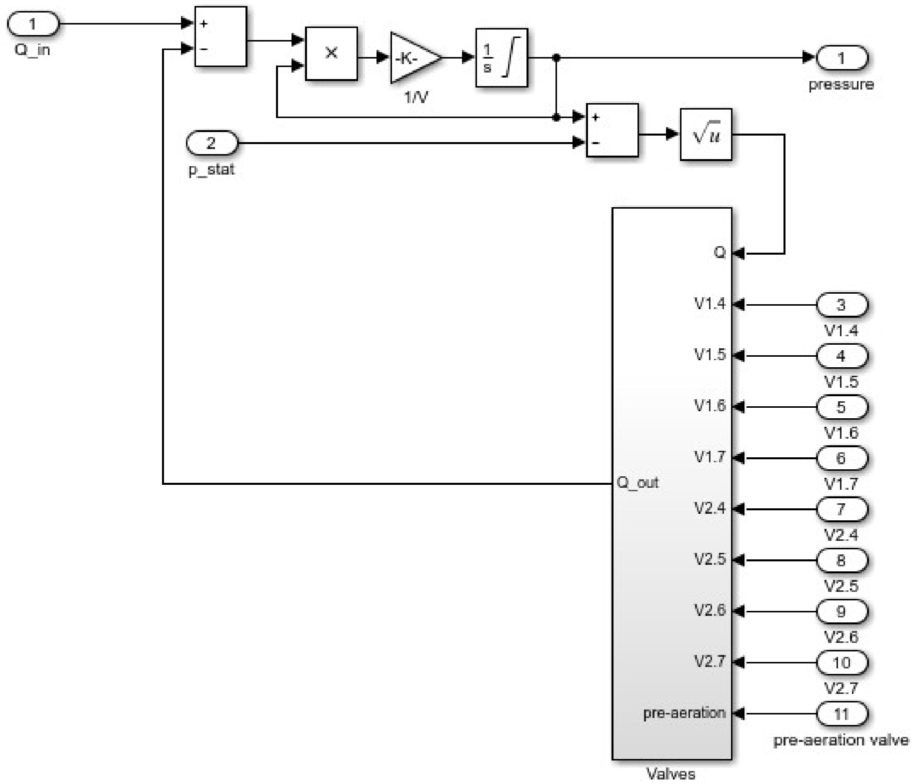

Figure 29. The model of the aeration distribution system.

Within the *'valves'* block, there are lookup tables for each valve that output K_v values corresponding to different valve positions. These are then used for obtaining the Q_{out} and further the change in the pressure. For both lines, section 6 and 7 have different valves than section 4 and 5. The inherent characteristics curves of the valves are shown in the figure 30. If compared to the study by Harja et al. (2016), this part of the model is quite different, as the k, is essentially gained directly through the actual total K_v of the control valves, rather than determining the k_i for each section experimentally. Furthermore, this study uses the actual characteristics of the valves, rather than assuming the linear characteristics used in Harja et al. (2016)

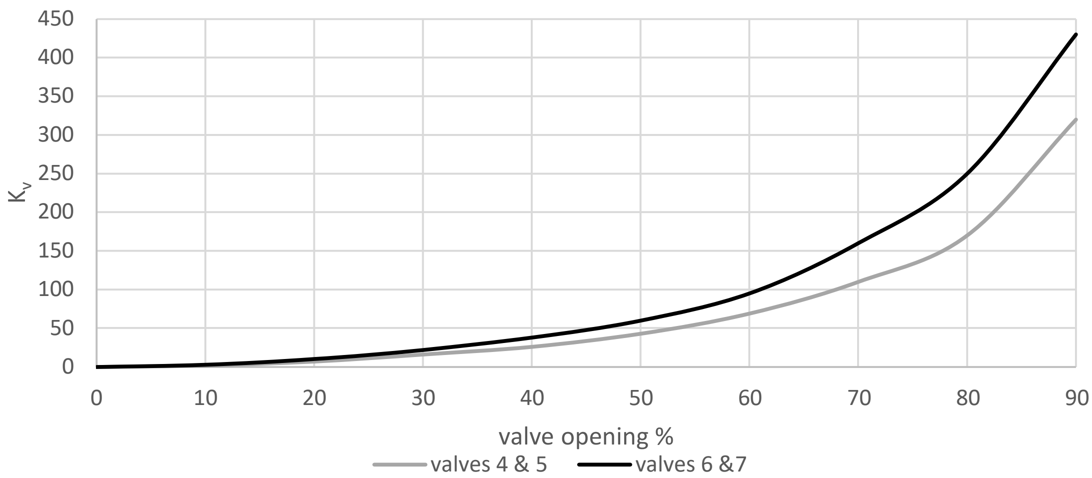

Figure 30. Inherent characteristic curves of the control valves.

5.5 Model calibration and validation

The model developed in this study takes only the control valve positions as inputs and adjust the flow rate to keep the system pressure constant. The model outputs the power, air flow rate, speed and discharge pressure of the blowers. Additionally, the number of starts required by the secondary blower is logged. Simplified flow chart presenting the process of calculating the values for each timestep is shown in figure 31.

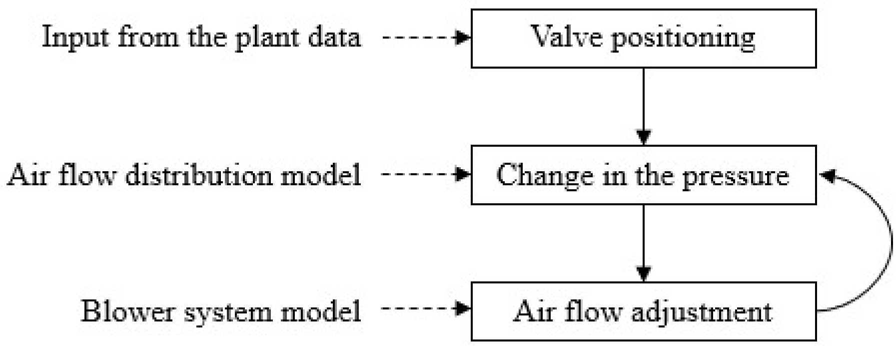

Figure 31. Process schematic of calculations during one timestep of the model.

Model has multiple adjustable parameters. The blower system has delays and thresholds for start and stops signals and the PI controller parameters for the pressure control of the blower. For the aeration distribution system, the final version of the model only included the offset for the valve block and the maximum position of the pre-aeration valve. Additionally, the lookup table values for the individual valves could have been modified, as it is likely that the installed characteristics of the control valves will differ from the inherent characteristics stated by the manufacturer. This was not done as it would have further complicated the model and made the calibration really challenging.

During the model calibration, a reference for the air flow rate was estimated by using the blower model and the known blower loadings. With this air flow rate, estimate for the k, under which the pressure would remain close to its setpoint, was solved. Because there are delays related to the starting and stopping of the secondary blower, it is clear that the pressure cannot remain perfectly constant. Nevertheless, this estimate for the k gives a scale for the values that are required from the valve block.

While using only the inherent characteristics of the control valve, the valve block yielded very similar behavior as the reference k, apart from some offset, during low load periods where only a single blower was running. Issues with the valve block output would arise during periods where two blowers would be running simultaneously. During these periods, the valve block would not yield high enough value for k, just by using the control valve positions. Also, the data from the plant indicated that during the operation of two blowers, the pressure would remain at its setpoint, as the blower loadings and valve positions would quickly stabilize during the startup of the secondary blower. Although the installed characteristics of the control valves are likely to differ from the ones stated by the manufacturer, the difference would not be enough to explain the differences between the high and low load periods. It may be that this behavior during higher loadings is influenced by the pre-aeration, for which the air is supplied by the same blowers. There is no data available for the pre-aerations control valve position, other than that its minimum position is 30% open. Because of the lack of data and unknown control strategy of the pre-aeration valve, it is assumed that under normal operation, the valve would operate throughout its range, so its position is estimated to increase during higher loadings. During the calibration, setting the pre-aeration valve to open steadily from 30-75% open during the startup of the secondary blower, yielded very good results improving the model's performance during the high loading periods.

Although the control parameters stated that the start and stops limits for secondary blower are 97% and 56%, of the primary blower load, these values where changed during the calibration to 99% and 55%, based on the model's performance and reference data. The delays were adjusted so that the primary blowers loading would have to remain over the threshold continuously for 9 minutes for starting the secondary blower and the loading would need to be below the threshold for stopping for 5 minutes. The minimum run time for a started secondary blower was set to 30 minutes. This would together with the delays prevent the unrealistic starting and stopping of the secondary blower.

For the calibration and validation, 5-day periods were used, and the model's performance was assessed based on performance of individual blowers and total blower power. For calibration, the data was from 6.-10.4.2019 and for validation, data was from 13.-17.4.2019. The normalized mean square errors (NMSE) for primary, secondary and total blower powers are presented in the table 3 together with the average relative errors for the primary and total blower power. The NMSE values are calculated with goodnessOfFit function in Matlab, for which the perfect score would be 1. For the secondary blower, the relative error is not given, as during the standstill operation, the power is zero and the relative error becomes undefined.

Table 3. Model performance parameters.

	Calibration			Validation		
	Primary	Secondary	Total	Primary	Secondary	Total
NMSE	0.40	0.63	0.84	0.44	0.77	0.86
Relative error	7.02%		5.00%	5.40%		5.34%

The error during the calibration and validation is acceptable, considering that during the cyclic starting and stopping of the secondary blower, differing start and stop times between the actual and modelled system increase the error quite a bit as can be seen in the figure 32 which is the total measured and modelled power and the relative error during the first day of the calibration period. From the figure 32, it is clear that otherwise small error can significantly increase, if the modelled power of the primary blower differs slightly from the measured value near the threshold values for starting and stopping of the secondary blower. It also should be noted that all the values, including errors, are observed in a one-minute scale. If hourly averages would be used, the effects of cyclic operation of two blowers would filter out, but as the purpose is to model the actual operation of the blowers, hourly averages would not represent realistic operation. This is why hourly averages are not used for analyzing the simulation results.

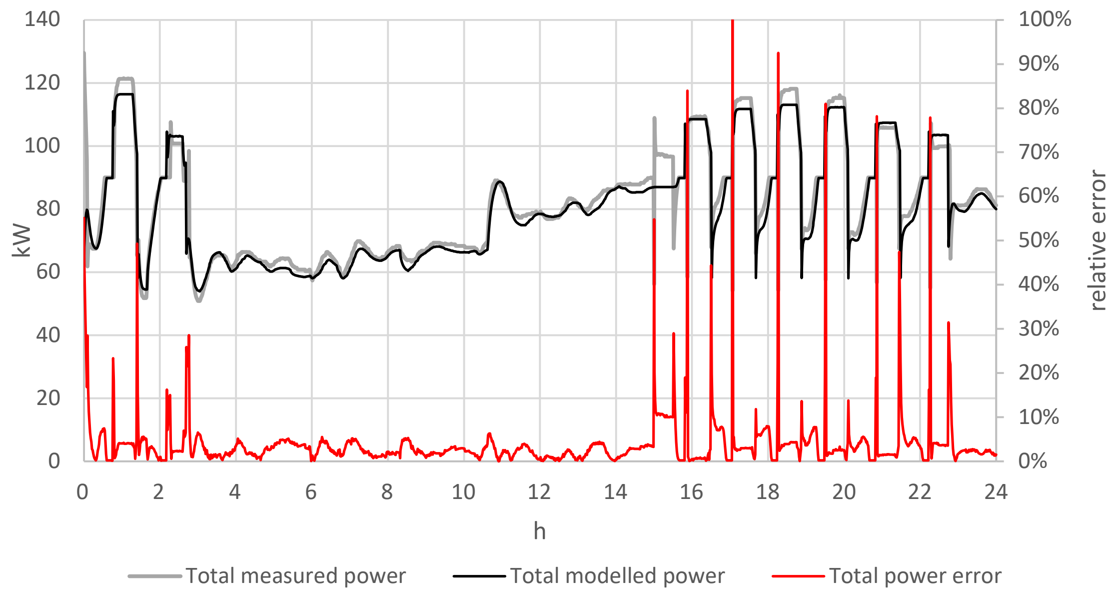

Figure 32. Total measured and modelled power and the relative error during the first day of the calibration period.

For assessing the frequency of required starts, the number of measured and modelled starts are compared. During the calibration period, the secondary blower is actually started 56 times, while the model required 53 starts. For the validation period, there are 34 actual and modelled starts, although all of the starts do not occur precisely at the same time. Two blowers are running simultaneously for 28.2% of the calibration period, while the model ran two blowers for 29.5% of the time. For validation periods the actual run time is 44.2%, while the model used two blowers for 49.0% of the total time. The total blower powers for calibration and validation are shown in the figures 33 and 34. Total powers are used for illustrative purposes, but in the further analysis, blowers with different duties are considered separately.

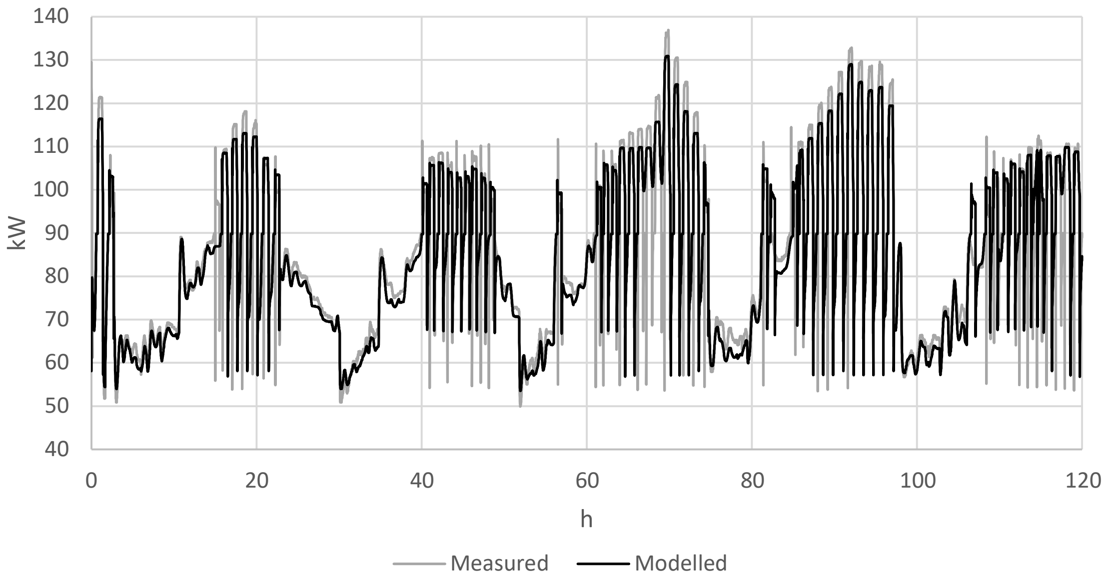

Figure 33. Measured and modelled total power during calibration period.

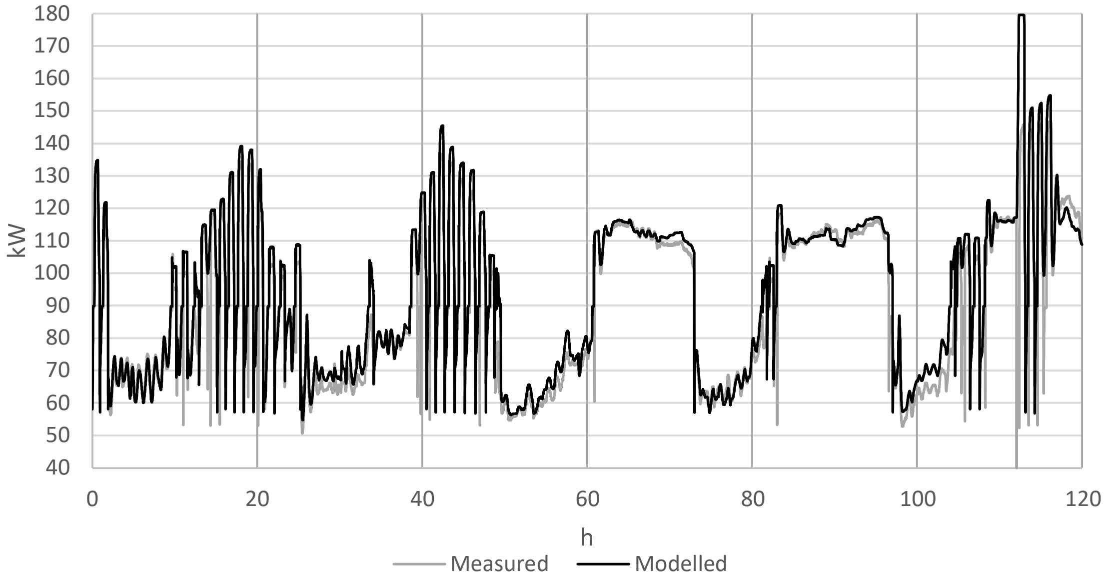

Figure 34. Measured and modelled total power during validation period.

During the calibration and validation, the system pressure remains stable, the average relative error being 0.16% during both calibration and validation periods. The pressure remains within 2 kPa 93% of the time during both periods, although there are some sharp peaks in the pressure as the secondary blower is started and stopped. Pressure also may remain below setpoint during times that the secondary blower is yet to start, while the primary blower is operated at its maximum point.

6 Blower load profiles and operating characteristics

6.1 Distribution of blower load points

For studying the typical operation of the high-speed turbo blower, 156-day simulation from the 4.12.2018 to 8.5.2019 was ran using the model of the Hermanninsaari WWTPs aeration system. For assessing the typical operating characteristics of the blowers, frequency distributions and typical load profiles are used. The frequency distribution of the total blower loads is shown in the figure 35.

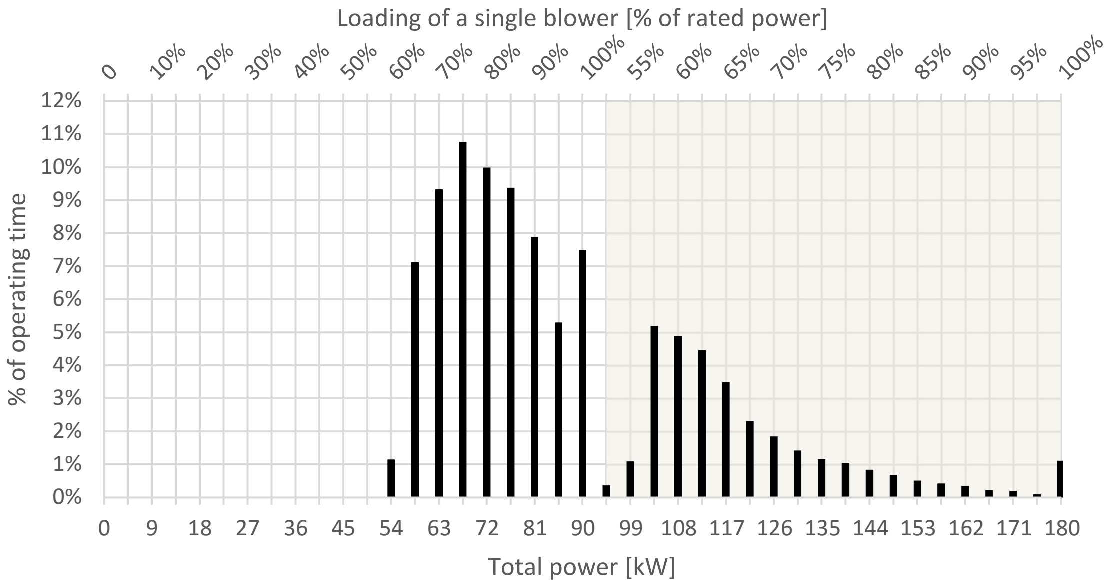

Figure 35. The frequency distribution of the total blower power. On top x-axis, corresponding loading for a single blower is shown and the area where two blowers are required is further highlighted with grey.

From the figure 35 it can be seen that most of the operating points fall under the single blower operation. During the simulated period, 68% of the operating time, only the primary blower is used. A depression can be seen in the middle of the frequency distribution. This is an area where two blowers are required, but the loading of a single blower is not enough for the control system to keep the secondary blower running. Also, these points are near the maximum turndown of the blowers where the operating point will be near the surge line. Systems air flow rate vs. blower loading is shown in the figure 36, which further illustrates the effect of starting the secondary blower.

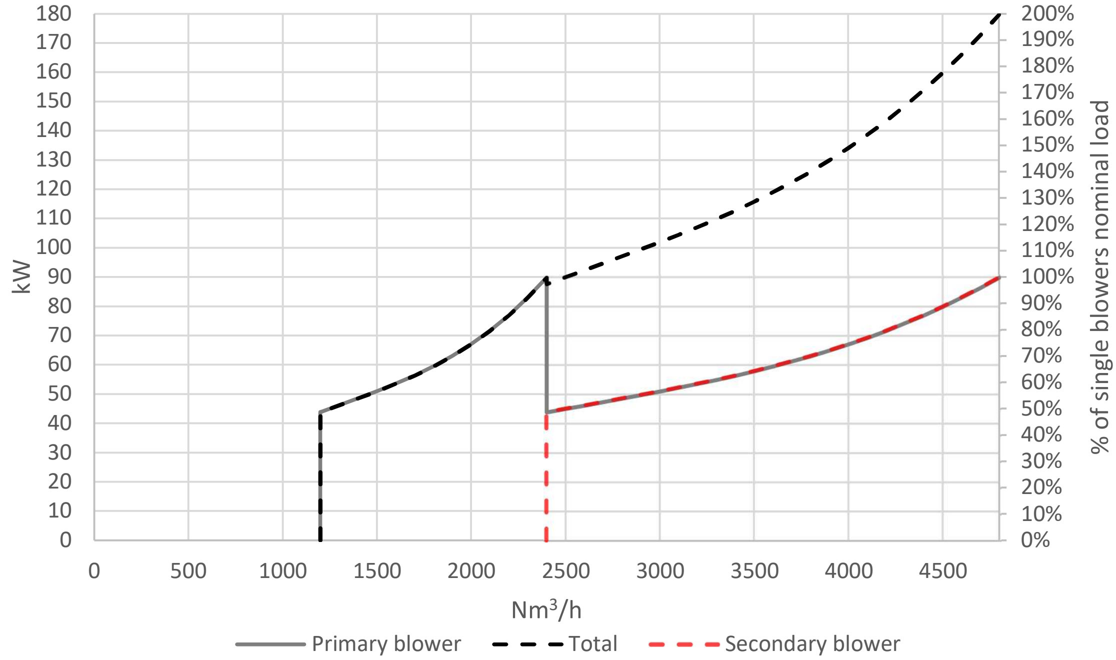

Figure 36. Aeration systems air flow rate vs. blower loading.

For the frequency distribution of a single blower, the cycle between different duties needs to be accounted for, as the simulation used only two blowers, which both remained at their duties as a primary and a secondary blower. As a single blower is cycled through three different duties, the frequency at a particular load point will be one third of each duty's frequency at that point. The frequency distribution of the load for a single blower is shown in the figure 37.

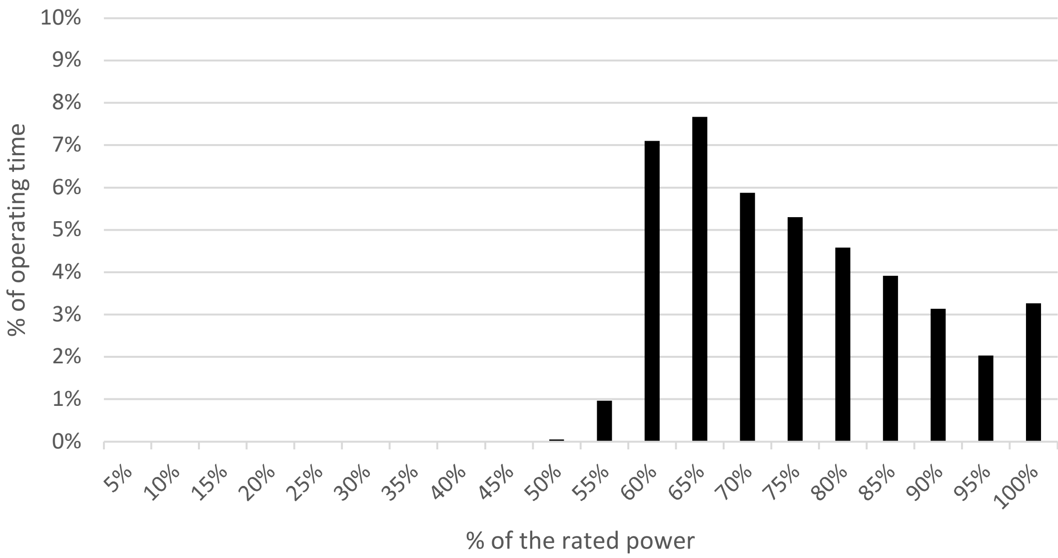

Figure 37. The frequency distribution of the loading for a single blower. Cycling duties as primary and secondary blower and standstill operation are considered, but 0% load point is not included in the figure for illustrative reasons.

From the figure 37, it can be seen that due to standby duty and the large portion of time spent at standstill during the duty as a secondary blower, single blower will be at rest 56% of the time. Also, the simultaneous operation of primary and secondary blower decreases portion of time near the nominal load, leading to that apart from remaining at standstill, blowers will be operated mainly under partial loads.

6.2 Operating patterns

Because of the diurnal variation in the plants loading, blower load profiles during different days will follow similar patterns. During low influent loading, only primary blower is operated, and the loading can vary throughout blowers operating range. As the influent NH_4^+-N increases, aerated volume will be increased by opening the valves of zones 4 and 5. With increasing aeration demand, eventually secondary blower needs to be started and whether the operation will be cyclic or not depends on the control valve openings of the zones 6 and 7, as the rest of the valves will remain at fixed positions. Eventually the influent loading decreases and aerated volume is decreased, and the system eventually goes back into operating only the primary blower. An example of the diurnal pattern of measured NH_4^+-N loading and modelled operation of the blowers is shown in the figure 38.

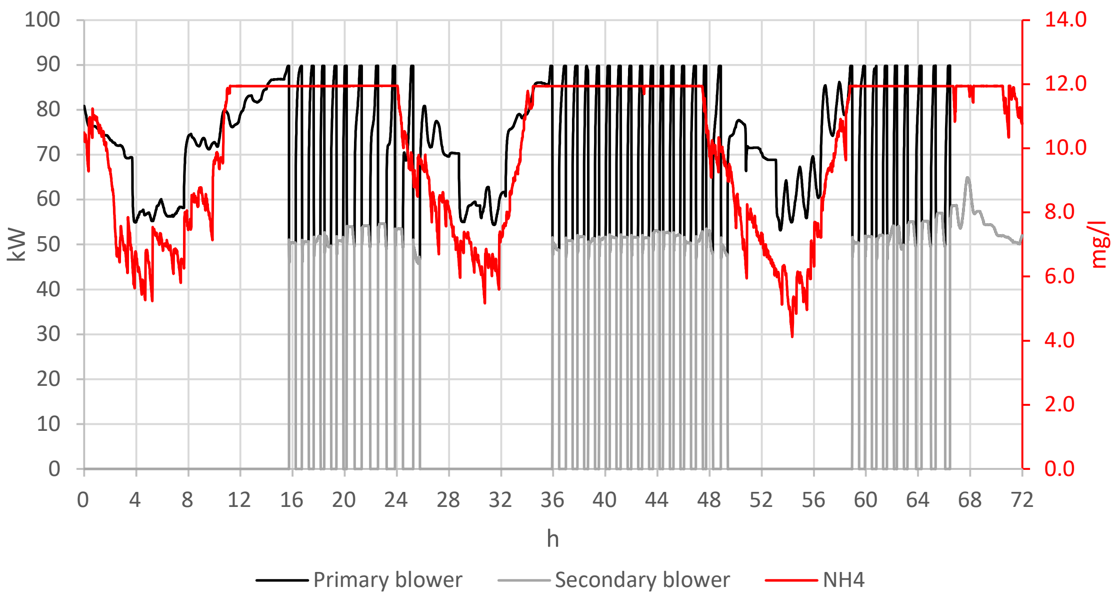

Figure 38. An example of the diurnal pattern of measured influent loading and modelled operation of the blowers.

From the simulated data, the operation of the blowers can be divided into three modes of operation: continuous operation of a single blower, continuous operation of two blowers and cyclic operation of two blowers. Largest portion of the total operating hours is continuous operation of single blower, which is during the low influent loads at nighttime and before noon. If we define the continuous operation of single blower to be operation without starting the secondary blower for at least 60 minutes, this mode of operation last for 12 hours on average. This is in line with the fact that the low influent load condition, with the reduced aerated volume, last on average for 11 hours, taking place usually between midnight and noon. Second most common mode of operation is the cyclic starting and stopping of the secondary blower. On average, the secondary blower is started 6 times per day. Of all the starts of the secondary blower, 88% lead to running the blower for less than 1 hour and 44% of the starts for the minimum run time of 30 minutes. Rest of the time is the continuous operation of two blowers. The portions of different modes of operation from the total operating hours is shown in the figure 39.

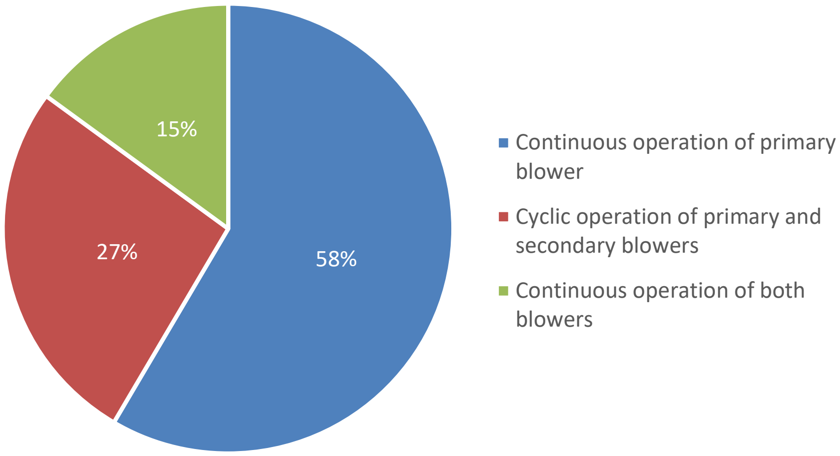

Figure 39. Portions of different modes of operation from total operating hours.

6.3 Blower speed range

Blowers will operate with relatively narrow speed range, roughly within 15% from the maximum speed limited by the power limit. This is because of the affinity laws that govern the operation of centrifugal blowers, and the fact that with a limited pressure range of a pressure-controlled system, the small changes in the blower speed will yield large changes in the blowers air flow rate (Dickherber 2010). The distribution of the simulated speed together with the average loadings of the primary blower is shown in the figure 40.

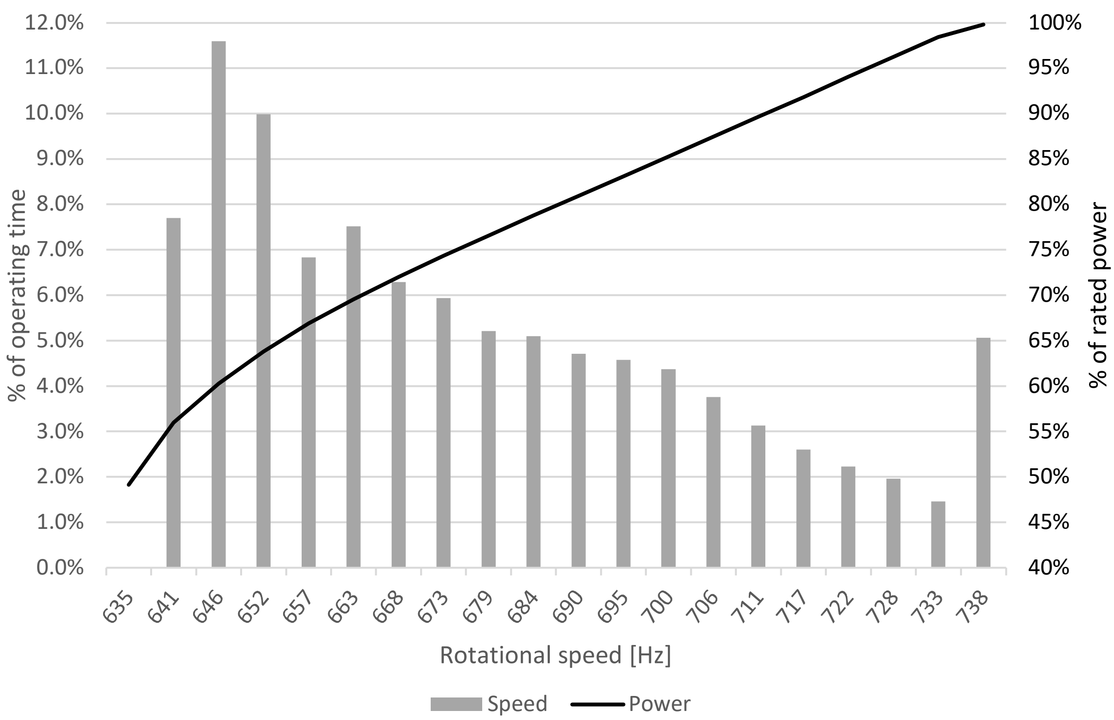

Figure 40. Distribution of blower speed and average loading with corresponding speeds with 91 kPa discharge pressure.

In the figure 40, the weight of the distribution is towards the minimum of the used speed range, but there is a slight increase at the point of maximum speed, because the primary blower will always operate near the power limit at a near constant speed, before the secondary blower is started. Reason why the distribution slopes downwards, is that as the secondary blower is started, the blower will decelerate as the load is divided between the two blowers. This leads to that during higher influent loadings and associated cyclic operation of the blowers, primary blower alternates between its speed at the power limit and the speed at the much lower load point during simultaneous operation of two blowers.

For the speed control, the model also calculates maximum and minimum speed based on the power and surge limit under varying backpressure. These limits set the usable speed range for the control and as the pressure increases the range gets narrower as upper and lower limit for speed start to approach each other. This can be seen from the previously presented figure 18, in the section 3.1.2. During the startup of the secondary blower, blower will be operated relatively near the surge line, as close to minimum flow rate is required from two blowers that are operated simultaneously. From the total operating hours of simultaneous operation of two blowers, 49% of the operation is within 10 Hz of the surge line. During the initial startup, 59% of the time is within 10 Hz of the surge line. There would still be room to operate the blowers with a lower speed, but as the control valves remain at their fixed positions during the startup of the secondary blower, the pressure setpoint will be met with a higher air flow rate than the process would actually require. Also, the limit for minimum speed used was based on the surge avoidance line, so there might be a possibility to decrease the flow rate even further.

6.4 Effect of reduced backpressure

The long simulation was also ran using 2 kPa lower pressure setpoint, to estimate the significance of the backpressure. The 2.2% reduction in the backpressure yielded 15% reduction in consumed energy. This experiment was purely speculative, as the valve positions used remained the same, meaning that the air flow requirement would need to be decreased in order for reaching the same DO levels. Because the control valves usually have room for opening even more, reducing backpressure might be possible through optimizing the valve control. Also, as DO setpoint is the same for all the controlled tanks, decreasing aeration requirement might be possible by optimizing the DO profile of the aerated tanks.

For further studying the possibilities to reduce the pressure setpoint and consequently the energy consumption, model should be expanded with a biological model, to ensure that the DO levels would remain sufficient with the reduced pressure setpoint. This again would increase the complexity of the model and reduce the length of the feasible simulation periods. The prospects for further studies regarding the system level optimization are discussed in the section 7.2.

7 Discussion

7.1 Blower operating characteristics and equipment level optimization

From the typical blower loadings, it is apparent that most of the operating hours, the blowers are operated under partial loads. Also, the operating hours under different partial loads are distributed quite evenly, so there are no clear operating points to which the efficiency of the blower's electrical equipment should be optimized. Also, as the time scale of the start and stop sequences does not necessary allow dimensioning of the equipment for cyclic duty, the motors and VFD will most likely be oversized for the application. It should also be noted that although the results from the simulations are valid for this particular system that is operating with a specific pressure setpoint, different systems may operate blowers with different loadings. Some common characteristics for the aeration blowers can be assumed, which are discussed in the following subsections.

7.1.1 Diurnal operating pattern

Operation will quite clearly follow diurnal pattern of the influent loading and has clear periods of high and low loads. The diurnal pattern is generally observed with the WWTPs treating municipal wastewaters and Leu et al. (2008) observed quite similar pattern in the air flow requirements, as was found in the simulation results of this study. Leu et al. (2008) also notes that as the diffuser efficiency decreases with higher air flow rates, the increase in the air flow requirement is pronounced during higher influent loadings. Henze (2008) states that the NH_4^+-N concentration often shows a diurnal pattern, as its main source is urine. In the case of Hermanninsaari WWTP, which treats municipal wastewaters and furthermore controls its aeration based on the NH_4^+-N concentration, it is clear that also the blowers will be subject to diurnal variation in the plants loading, due to human activity.

The different patterns identified in this study are realistic, as the cyclic hunting of the blowers and valves is a typical issue with automated DO controls (Henze 2008). Spellman (2013) also characterizes it as an issue of smaller pressure-controlled systems and suggests that directly controlling the air flow, without additional pressure control loop may solve the issues with control systems instability. Henze (2008), recommends blowers with wider turndown ranges and control systems that would acknowledge the limited capabilities of the blowers by including a blower model within the control system. In this system in particular, the cyclic operation seems to be related to the DO interlock time and operation close to a threshold in which additional blower would be needed, but after its start-up the air flow requirement is not actually high enough for the control system to keep the additional blower online.

During the start sequence of the secondary blower, there is an interlock time of DO control, which keeps the valves at constant position for 30 minutes as the secondary blower is started. From the simulation results, it was seen that the large portion of the total starts of the secondary blower, lead to running the blower for this minimum time. Most likely interlocking the DO control, while increasing the air flow rate leads to that the DO levels increase and as the valves go back into their modulating duty, they start to close down immediately. This in turn causes decrease in blower loadings and eventually the secondary blower is shut down. If it would be possible to decrease this interlocking time, periods with over aeration would be reduced and consequently the energy efficiency of the system would improve. This would likely increase the amount of starts required per day, but at least from the electrical equipment point of view, this should not be a problem, if the cycling would remain within reasonable limits. Another alternative to consider could be to increase the valve opening and reducing the pressure setpoint during the starting and stopping of the secondary blower. Although this might increase the air flow rate over the actual demand, the power drawn might decrease due to reduced backpressure.

Rohrbacher et al. (2014) studied a plant which operated two multistage units, one of which was operated continuously and other which was started during the periods of higher air flow requirement, similarly to the Hermanninsaari WWTP modelled in this study. Rohrbacher et al. (2014) estimated from plants operational data from January to March, that only a single blower was operated 84% of the total operating hours. The simulation results from the Hermanninsaari WWTP, also indicated that a significant portion of the total operating hours requires only a single blower. This is most likely true for most of the similar aeration systems, as the full capacity of the blower system is typically required for only a limited time period (Jenkins 2013).

7.1.2 Requirements for motor control

Simulation results support the statement by Halkosaari (2006), that there is no need for a fast speed or torque response from the motor control. This is because the process changes are relatively slow, largest changes being during the start and stop cycles of additional blower and during the increase of the aerated volume, which from the VFDs standpoint would still be happening quite slowly. Control strategy of the control valves highly influences the operation of the blowers and therefore systems energy consumption. This is clearly true for system, which use pressure control, as the blower's output will mainly depends on the valve positioning as explained in Åmand et al. (2013). This contributes to the low requirements for dynamic motor control during the normal operation, as the valves will adjust their positions quite slowly and even the large changes in the influent loading would result in relatively mild control actions from the blowers.

Considering operation near the surge conditions, precise torque control might be beneficial, as it could support the surge protection system of the blower by detecting impending surge from the motor torque. Dickherber (2010), suggests that air flow transmitter together with a low flow detection logic would provide improved protection and extended operating range for a blower, compared to for example low-amperage switch. If the surge detection function would be integrated within the VFD controls, there would be no need for the external flow transmitter and as Gravdahl et al. (2002) showed, it might be even possible to implement surge control through the VFD. This would in turn further highlight the requirement for more precise motor control, when surge detection and control functions would be implemented within the VFD. Precise motor control could also help with the startup of the additional blower, as it could ensure that the stable operation of parallel blowers is quickly reached, reducing the times when the process is overaerated because of the interlocking of the DO control.

7.1.3 Limited operating range

High-speed turbo blowers and centrifugal blowers in general have limited operating range due to surge condition, that limits the operation in the low flow region. Because of the physical limitations of centrifugal blowers, the typical operating range will be between 50 and 100% of the rated flow, although this might vary between installations (Spellman 2013). The simulation results showed that the blower will be operated within these limits, with more emphasis being on the lower end of the operating range, due to the control strategy of the blowers, in this particular system.

With a constant pressure system, the power limit for speed will not fluctuate that much and the used speed range of the blower will be quite narrow, because of the changes in the air flow rate will be quite large with relatively small adjustments in rotational speed. If the pressure setpoint would be different, the speed range would be shifted, but it would still remain quite narrow, although reduced pressure would allow wider operating range. From the VFDs standpoint, this would allow the focus on the higher end of the required output frequencies as the speeds below the blowers operating range are quickly passed during the startup of the blower. The limited range of blower speeds with a constant pressure system, is also noted by Dickherber (2010).

7.2 System level optimization

Backpressure has significant effect on the systems power consumption and even small reductions can yield large savings. If the operating characteristics of the system will not allow optimization of the electrical equipment of the blower, focus should be given to optimizing the process so that although the equipment would be operated outside its best efficiency range, total power requirements would be at their minimum. Some of the power-minimizing control strategies are the MOV-logic and direct flow control, which by definition should not affect the process control itself but would reduce the blower power requirements through minimizing the discharge pressure (Åmand et al. 2013). Vrečko et al. (2014), mentions that it is important to minimize the header pressure, as it ensures that the control valves will be at their maximum opening and therefore minimize the blower power. From the Hermanninsaari WWTPs process data presented in the appendix 1, it can be seen that the valves could be more open and there might be a possibility to optimize their control to minimize the pressure setpoint. Åmand et al. (2013) also mention that for a pressure-controlled system such as Hermanninsaari WWTP, alternative could be to relate the pressure setpoint to the total air flow requirement of the process.

For system operating only two blowers, there is not much room for optimizing their parallel operation for energy efficiency. In this particular system, there is a negligible range of air flows, for which operation of two blowers would be more efficient than operating only a single blower. For larger systems, where more blowers are required, there might be more possibilities to optimize the parallel operation, by mainly operating the blowers at their BEP and using a single blower for adjusting to process changes. This idea of operating only one blower in modulating duty is a part of compressed air systems best practices mentioned in Hall (2018), but for a smaller treatment plants, operating multiple blowers for only a limited time periods, this approach is not possible.

For studying different system level changes, such as altering the valve controls, the aeration system model should also include the biological model, so that the constraints of the process performance could be considered. With the expanded model, it is likely that the length of simulated period needs to be reduced but simulating operation during for example average day or significantly higher or lower influent loads could be feasible. The biological model could also offer possibility for studying the effect of altering the oxygen profile of the aerated sections of the plant and possibly help to reduce the air flow requirement of the process. It could also be possible to include this aeration system model as a part of a full-plant model, which would then allow inclusion of plants' other unit processes in the optimization. For comparing different blower technologies and configurations, the model could be used as it is, by simply replacing the model developed for the particular high-speed units, with a different type of blower. Additionally, different types of valves can be compared, by setting their characteristic curves into the model.

8 Conclusion

In this study, a model of diffused aeration system and high-speed turbo blowers was developed, to assess the typical operation of the blowers and for finding prospects for equipment level energy efficiency optimization. The distribution of the blower's loadings and typical operating patterns were identified for the Hermanninsaari WWTP, and the results supported the findings in the literature. The results were analyzed from a point of view of VFD utilized in high-speed turbo blower, for which the blower application in wastewater aeration was characterized.

The varying influent loadings and usually oversized systems make equipment level energy efficiency optimization challenging. For increasing the aerations energy efficiency, system level design has significant effects on the energy consumption. Smaller systems with few blowers have little room for optimization regarding the blower control, but precise motor control could be used for supporting surge detection and protection systems, making operation near surge condition safer, which also is desirable from energy efficiency perspective, as the surge limits the operation with low air flow rates and may reduce the possibilities to save energy during periods with low air flow requirements.

Aeration system and blowers can be modelled with sufficient accuracy, by using the polynomial approximation for blower curves and control valve characteristics. Model could be extended with biological model, to assess different valve control algorithms. Modelling process itself gives important insights on aeration systems characteristics and makes it possible to estimate systems energy requirements and reasons behind it.

High-speed technology provides energy efficient alternative with multiple fringe benefits, that make it a worthy candidate while selecting a blower for a wastewater aeration system. High-speed blowers have more electrical complexities than traditional technologies, but these complexities can be managed through the support from the manufacturers. Under normal operation, they are more hands-off equipment for the operator than the traditional blowers. For high-speed blowers VFD, high switching frequencies are required and often output filter needs to be utilized for ensuring the quality of the output waveform. As the energy efficiency optimization of the equipment for this particular application is challenging, focus on the VFDs side should be put on the motor control performance and equipment's reliability, to ensure the blowers operation under varying conditions.

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List of appendices

Appendix 1: The process data of Hermanninsaari WWTP from 4.12.2018 to 8.5.2019

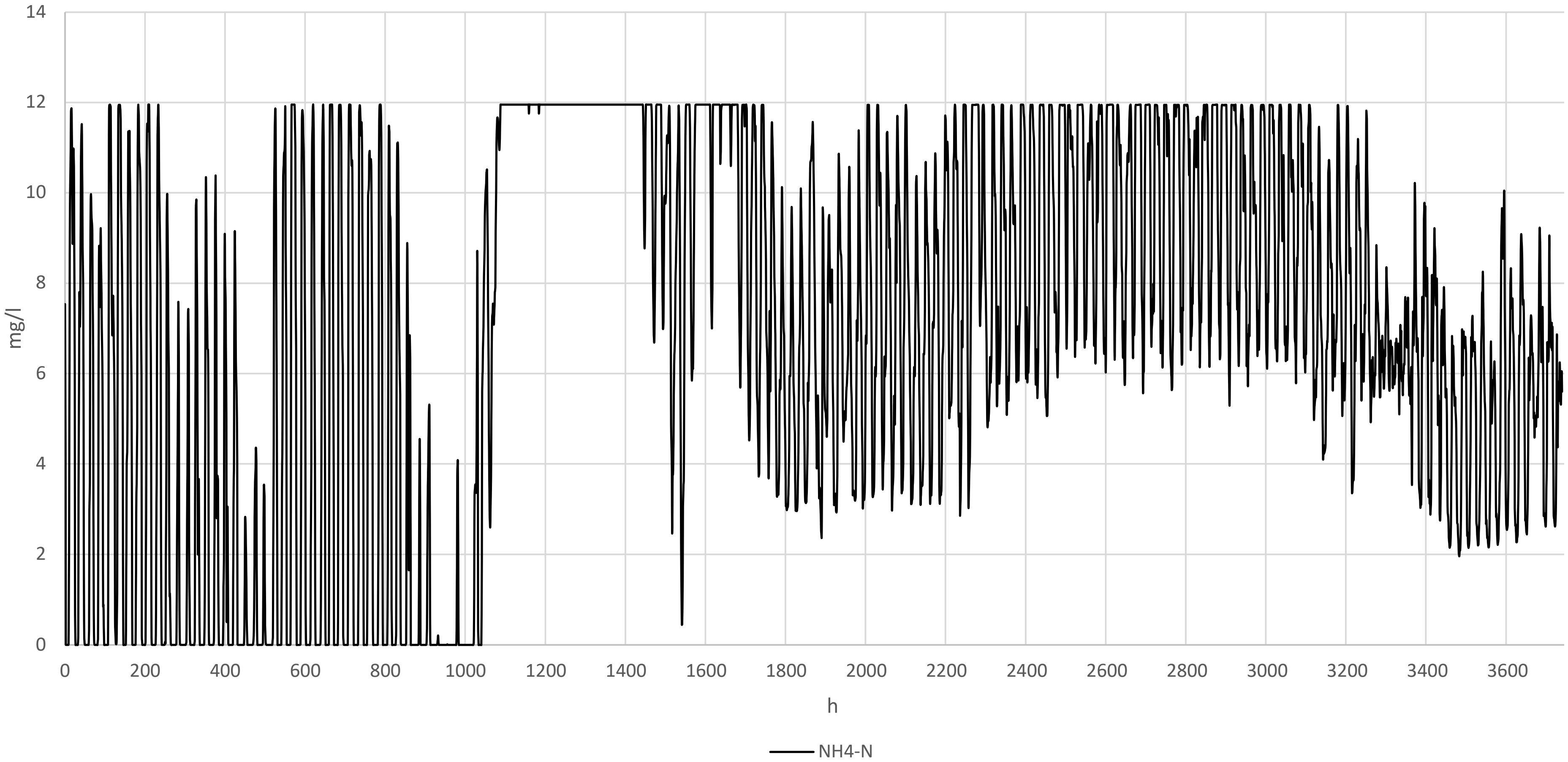

Figure 1. Hourly average of NH_4^+-N measurement

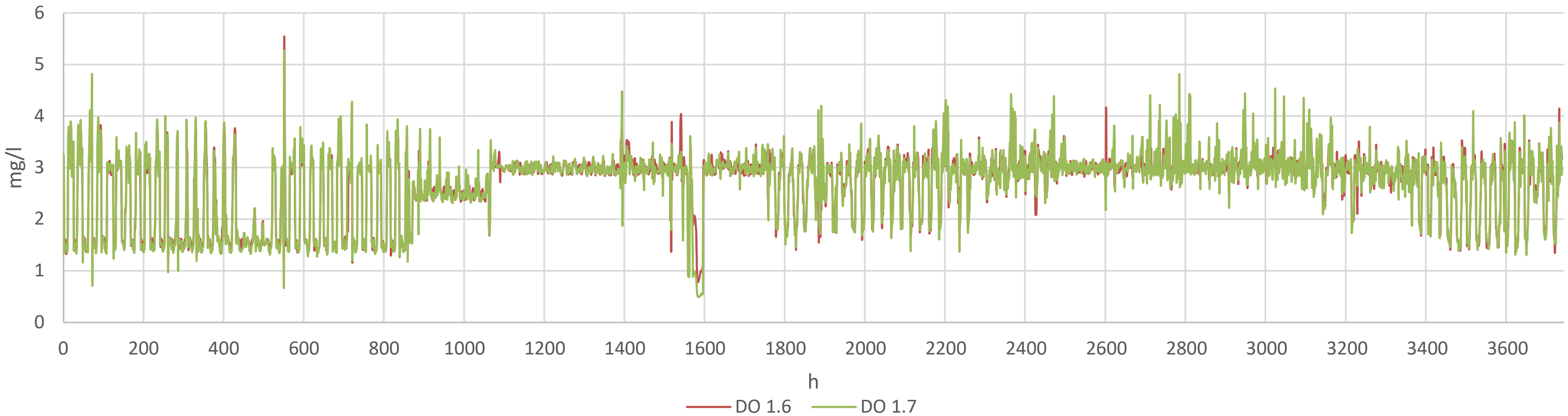

Figure 2. Hourly averages of dissolved oxygen measurements in zones 6 and 7 of treatment line 1.

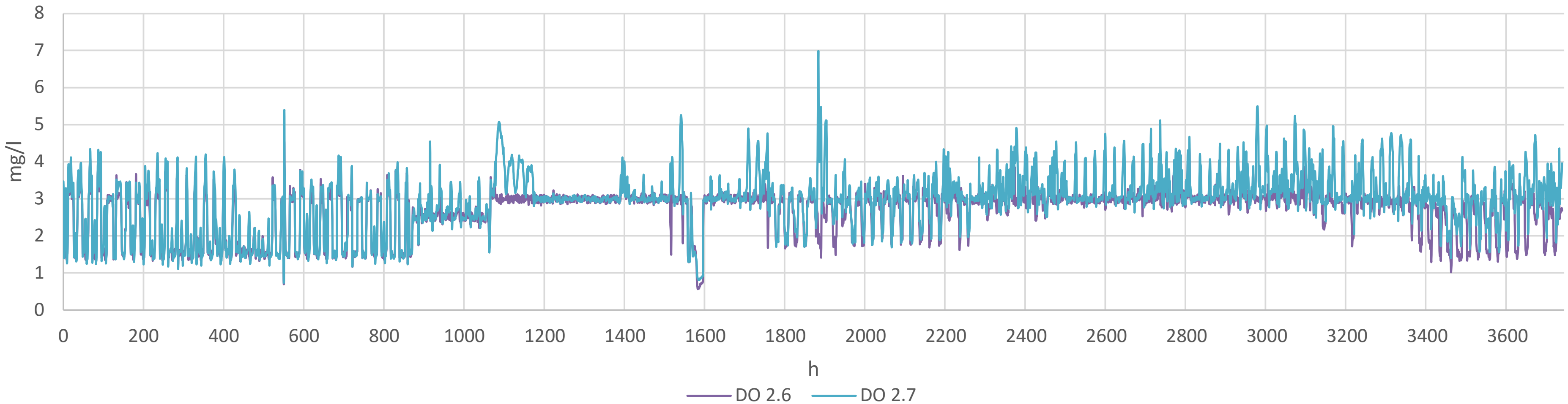

Figure 3. Hourly averages of dissolved oxygen measurements in zones 6 and 7 of treatment line 2.

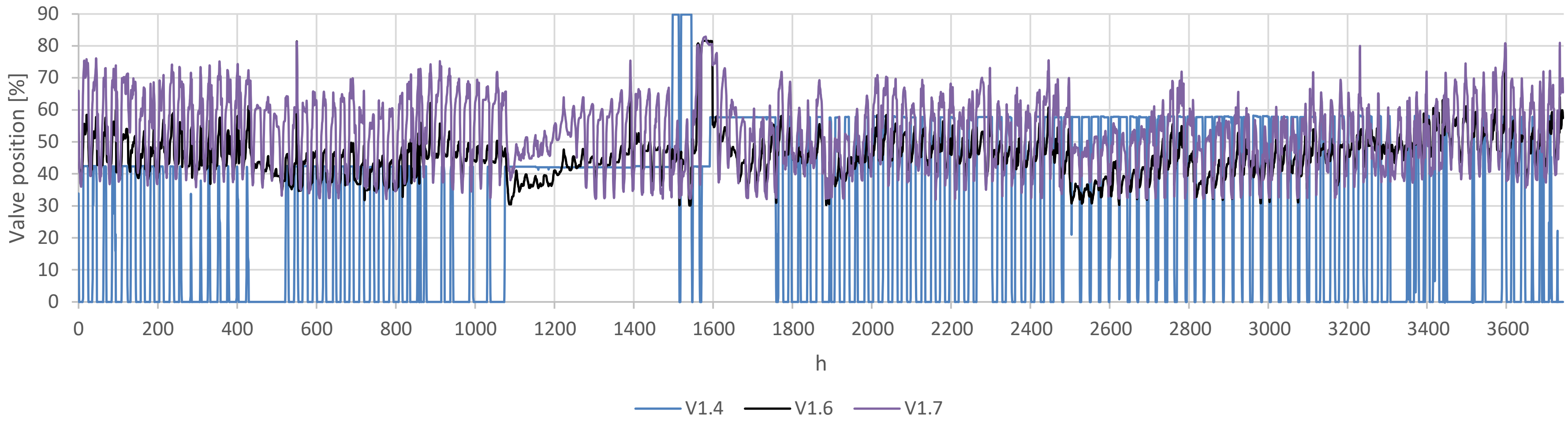

Figure 4. Hourly averages of the control valve positions in treatment line 1. Control valve in zone 5 was not in operation.

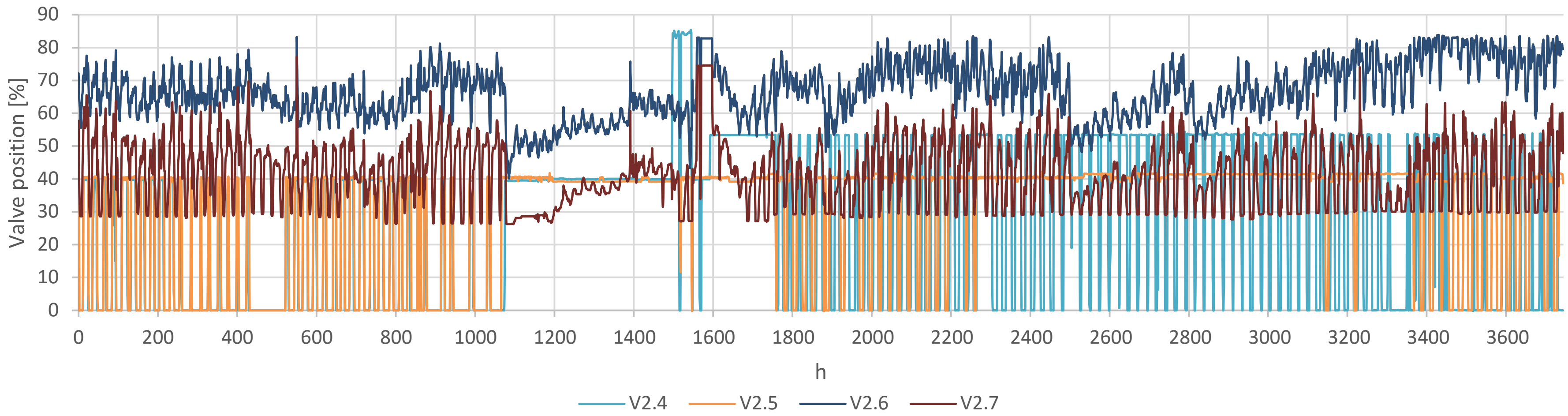

Figure 5. Hourly averages of the control valve positions in treatment line 2.